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RESEARCH ON DRY-TYPE COOLING TOWERS FOR THERMAL ELECTRIC GENERATION

Part I

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27 Abstract

An economic analysis is made for the use of dry cooling towers in thermal power plants in the United States. Twenty-seven sites were examined providing in each case capital and operating cost for natural and mechanical draft systems both for fossil and nuclear plants.

System optimization was based on capital cost, auxiliary power cost, cost due to loss of capacity, and fuel cost.

Comparison was made with wet cooling tower systems. ~~It was found that~~ with all factors considered, dry towers would be economically competitive with wet cooling tower systems. (Shirazi-EPA)

~~This report was submitted in fulfillment of Contract No. 14-12-823 under the sponsorship of the Federal Water Quality Administration.~~ ~~(Shirazi-EPA)~~

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RESEARCH ON DRY-TYPE COOLING TOWERS FOR THERMAL ELECTRIC GENERATION: PART I

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FOREWORD

The production of electrical power requires that large amounts of waste heat from the generating process be rejected to a heat sink. The usual method of accomplishing heat rejection has been to use circulating water, either from a natural body of water (once-through system) or from an evaporative-type cooling tower or cooling lake, to carry away the waste heat. The use of a once-through system results in the addition of heat to the natural body of water. The use of an evaporative-type cooling tower or cooling lake results in the consumption of water to replace that lost by evaporation in the cooling process.

A method of waste heat rejection by means of air-cooled heat exchangers, which transfer heat directly to the atmosphere without addition of heat to natural bodies of water or evaporation loss of water, is available to the utility industry.

In this report, information is presented on the theory of dry cooling as it would apply to steam-electric generating plants; operating results are summarized for several existing dry cooling tower installations; the comments of various equipment manufacturers are summarized; and the results of economic analyses made for dry cooling systems are presented for 800-mw fossil-fueled and nuclear-fueled generating units for 27 representative sites in the United States reflecting a range of fixed-charge rates, fuel costs, and weather conditions.

Following is a summary of certain of the more important conclusions reached as a result of the study:

1. There is need for a method of disposing of waste heat from steam-electric generating plants which does not add heat to natural bodies of water or require large quantities of make-up water for evaporative-type cooling towers.

2. Steam-electric generating plants equipped with dry-type cooling systems which discharge waste heat directly to the atmosphere are in successful operation in Europe. Two small generating units in the United States are also equipped with dry-type cooling systems.

   In a number of such plants, dry-type cooling systems were selected either as a result of better economic evaluation as compared to evaporative-type cooling, or because of an insufficient make-up water supply for an evaporative-type cooling tower.

3. As a result of technology and experience gained with air-cooled heat exchangers in industry, United States manufacturers can design and produce such dry-type cooling towers for generating plants. Air-cooled heat exchangers are now commonly used in the petroleum refining industry, and in petro-chemical and chemical process plants. Such air-cooled plants have been built and are in operation with heat rejection loads equivalent to large steam-electric generating plants. A number of plants dissipate up to 2 billion Btu per hour, a heat rejection load equivalent to a 425-mw generating plant.

4. The performance of a dry-type cooling system is measured by the temperature difference between the condensing steam of the turbine exhaust and the ambient air entering the cooling coils (called "initial temperature difference", or ITD) required to reject the design heat load.

   The capital cost of a dry-type cooling system increases with decreasing ITD; i.e., the capital cost of a 40°F ITD system will be higher than the capital cost of a 60°F ITD system for the same heat rejection load. Conversely, more efficient turbine operation will be obtained with the lower ITD (more expensive system).

5. A generating plant equipped with a dry-type cooling system of optimum economic size will experience some loss of generating capability as a result of increased turbine back pressure during hot weather.

   In this report, it was assumed that such lost capacity was replaced by means of peaking plants for a capital cost of $100 per kw. Other methods of restoring capacity of fossil-fueled plants are available including: removing feedwater heaters from service; use of a second steam admission point on the turbine with increased boiler capacity; and use of over-pressure throttle steam. Because of reactor licensing limitations, such methods would not apply to nuclear plants.

6. Turbine manufacturers are currently performing research on a new line of utility turbines especially designed for high back-pressure operation and are also studying the feasibility of modifying present designs to operate at the

high back pressures that will be encountered with dry-type cooling tower operation.

7. For a typical 800-mw generating plant in the Chicago area, the capital cost (based on 1970 price and wage levels), ITD, loss of capability during hot weather and cost of replacing such lost capability for economically optimized dry-type cooling systems are estimated to be as follows:

Fossil-Fueled Plant

Type of tower:	Mechanical Draft	Natural Draft
Capital cost, $/kw	$17	$20
Initial temperature difference ..	60°F	56°F
% loss of capability during hot weather	7.6%	6.4%
Penalty for loss of capability at $100/kw replacement* ...	$ 8	$ 6
Capital cost of dry tower system plus replacement peaking capacity	$25	$26

Nuclear-Fueled Plant

Type of tower:	Mechanical Draft	Natural Draft
Capital cost, $/kw	$23	$27
Initial temperature difference ..	65°F	62°F
% loss of capability during hot weather	13.6%	12.5%
Penalty for loss of capability at $100/kw replacement* ...	$14	$13
Capital cost of dry tower system plus replacement peaking capacity	$37	$40

*On the basis of 800-mw capacity (capital cost of peaking capacity required to restore lost capability during hot weather, $ divided by 800,000 kw).

The foregoing analysis is on the basis of 5,250,000 mwh annual production (75 percent plant factor).

8. The use of dry-type cooling systems with steam-electric generating plants will eliminate the need for a large supply of water as a basic site requirement and will result in greater freedom of plant siting than has been possible.

9. There are large deposits of coal and lignite in the United States which are not yet fully developed—notably in Arizona, Montana, North Dakota, Utah and Wyoming—which lack sufficient local water supplies for the make-up requirements of evaporative cooling means. Except for the use of dry-type cooling systems, the alternatives available for development of these coal and lignite supplies for large generating plants are to bring water to the mine-mouth plant sites or to transport the fuel to a plant site where water is available.

The use of dry-type cooling systems with mine-mouth generating plants in these areas opens up new possibilities for use of the important fuel reserves.

10. The results of the economic studies made in this report indicate that the total bus-bar power costs of a typical large fossil-fueled generating plant equipped with a dry tower cooling system will be approximately 0.48 mills per kwh higher than the total bus-bar cost, including fixed charges, of a similar plant equipped with an evaporative-type cooling tower, a difference of approximately 7 to 10 percent. When considered at retail level for residential service with all costs of generation, transmission and distribution reflected, the increase in cost for the dry-type production will be about 2 to 5 percent, depending upon rates. A 2 to 5 percent increase in a $20 residential monthly electric bill is equivalent to 40¢ to $1.00, and an increase of even this amount would not occur unless all generating plants in a utility system are cooled by a dry-type system. For industrial power service, the increase would be approximately from 2 to 6 percent.

There are a number of possible savings available to a utility with dry-type cooling systems which would tend to reduce or possibly offset the increased production costs from a dry-type cooling system:

a. Possible fuel cost savings as a result of the greater flexibility of plant location with a dry-type cooling system. A savings of approximately 5¢ per million Btu in fuel would entirely offset the cost difference of approximately 0.48 mills per kwh estimated above.

b. Possible transmission cost savings as a result of greater flexibility of plant location.

c. Possible savings as a result of the economies of an additional unit at an existing facility where inadequate water supply would otherwise rule out the addition.

d. Possible savings in cooling water make-up when compared to an evaporative-type cooling tower plant. For a cooling water make-up cost of $100 per acre foot, the water savings for the dry tower installation would approximate 0.2 mills per kwh.

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SECTION I

INTRODUCTION

Purpose of Report

The purpose of this report is to present the results of research conducted by R. W. Beck and Associates in connection with the use of dry-type cooling towers with steam-electric generating plants. Dry-type cooling towers transfer the heat of condensation of the turbine exhaust steam to the atmosphere by means of air-cooled heat exchangers with no evaporation loss of circulating water to the atmosphere.

Because of the growing shortage of large volumes of water for industrial and power generation cooling services, the concern with the effects of adding heat to natural bodies of water and the consumptive use of water with evaporative-type cooling towers, it is important to have available in one publication a source of technical information covering the present state of the art of dry-type cooling towers. The dry-type tower has no consumptive use of water by evaporation, nor does it require that water of high salinity content be drained off from the cooling water cycle and wasted, as is the case with the conventional evaporative cooling tower.

Nearly all of the technology associated with the dry-type tower for steam-electric generating plants has been developed in Europe. However, United States manufacturers have adequate know-how and experience in the design and construction of liquid-to-gas heat exchangers in industry, especially in chemical and refinery processes, to design and produce dry cooling towers.

There are a number of steam-electric generating plants in successful operation in Europe with dry-type towers, but to date only two small dry tower generating units have been constructed in the United States. The largest is a 20.18-mw, nameplate capacity, generating unit of the Neil Simpson Plant of the Black Hills Power and Light Company at Wyodak, Wyoming, placed into service in 1969. This was preceded by a 3-mw unit installed in 1962 at the same location.

Heat Rejection in Power Production

The production of electrical power requires that enormous amounts of waste heat be rejected. In the case of the conventional fossil-fired steam-electric generating unit, the waste heat is rejected partly to the atmosphere in the form of products of combustion from the steam-generating equipment, but the larger part is rejected to the cooling-water circuit.

By far the greatest heat rejection is from the main steam condenser. Other minor heat rejections are from the generator I^2R losses and the mechanical losses from the turbine and auxiliary rotating equipment. For a modern fossil-fired plant, approximately 4,800 Btu are rejected to the circulating water for each kwh of energy produced.

With a pressurized-water or boiling-water nuclear generating plant, the heat rejection to the circulating water is approximately 50 percent greater than for a fossil-fueled plant. However, the use of the high-temperature gas-cooled reactor nuclear plants will result in waste heat rejections to the circulating water which are comparable to those experienced by fossil-fueled generating plants.

When it is realized that the heat rejection to the circulating water from a modern fossil-fueled plant is equivalent to approximately half of the fuel burned in the boiler, and the heat rejection from a typical nuclear plant amounts to approximately two-thirds of the nuclear heat generated, one can appreciate the enormity of the thermal problem.

Existing and Estimated Future Power Generating Capacity and Requirements in the United States

The National Power Survey (1), a report written by the Federal Power Commission, has projected that the electrical-energy requirements of the United States will increase from 1.6 trillion kwh in 1970 to 2.8 trillion kwh in 1980—a 75 percent increase in 10 years. The report also predicts that by the year 1980, 87 percent of the energy will be generated by either fossil- or nuclear-fueled plants in the ratio of 68 percent fossil fuel to 19 percent nuclear fuel. At the present, either once-through condenser cooling (using natural bodies of water) or evaporative-type cooling towers are used for generating station heat dissipation. However, presently unused water, formerly available for evaporative-type cooling purposes, is rapidly decreasing because of other higher priority uses. In addition, large blocks of power generation are creating increasingly undesirable thermal pollution problems in once-through condenser cooling installations. Future increases in power generation will place a great strain upon our available water supply. Therefore, waste heat removal from the projected increase of generation will require that new methods be investigated for its disposal. Table 1 (2) illustrates the increase in peak demands and energy requirements from 1970 to 1990.

TABLE 1

Predicted Increase in Future Electrical Requirements

Region	Ratio of 1980 to 1970 Peak Demand	Ratio of 1980 to 1970 Energy	Ratio of 1990 to 1970 Peak Demand	Ratio of 1990 to 1970 Energy
Northeast	1.8	1.8	3.2	3.2
East Central	1.9	1.8	3.4	3.4
Southeast	2.1	2.2	4.1	4.1
West Central	2.0	2.0	3.8	3.8
South Central	2.3	2.3	4.5	4.7
West	2.0	2.0	4.0	4.0

The areas comprising the six National Power Survey regions are shown in Figure 1 (2).

Increased size of generating units and plants. Traditionally, the economics of capital construction costs and operating efficiency have resulted in a trend towards larger generating units. Construction costs per kw of unit capacity decrease with unit size, and plant labor requirements are more nearly proportional to machine units than to plant kw capacity. Although a small unit is not inherently less efficient than a large unit, the costs required for adding specific features resulting in higher net plant efficiencies are prohibitive for small units. Examples are high pressures, high temperatures, reheat, superheat and automated features. From a maximum size of approximately 200 mw at the end of World War II, the size of generating units ordered has increased to 1,300 mw, and utility industry leaders predict that, by 1990, units of 2,000-mw size will be in use.

In recent years, a number of electrical utilities have formed power pools in which the utilities join in the construction of generating units larger than any which the individual utilities could accommodate alone. Since large blocks of power can be transmitted long distances to load centers of widely separated pool members, the construction of extra-high-voltage transmission systems has contributed greatly to the feasibility of such large generating units. Table 2 illustrates the trend towards larger sized units and generating plants.

From the foregoing, we can only conclude that the problem of disposal of waste heat from steam-electric generating plants will become more acute in the future.

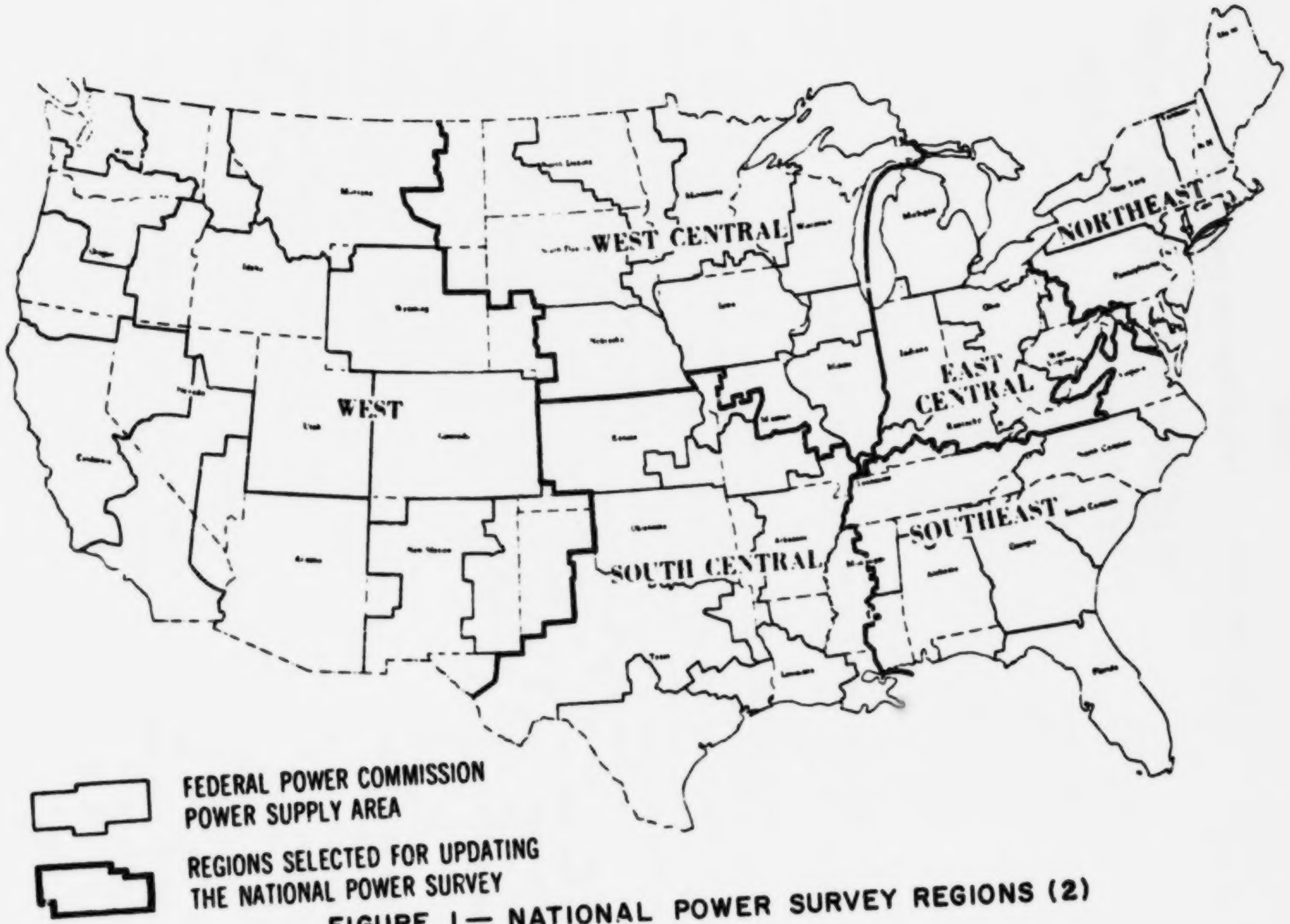

NORTHEAST
WEST CENTRAL
EAST CENTRAL
WEST
SOUTHEAST
SOUTH CENTRAL
FEDERAL POWER COMMISSION POWER SUPPLY AREA
REGIONS SELECTED FOR UPDATING THE NATIONAL POWER SURVEY

FIGURE 1 — NATIONAL POWER SURVEY REGIONS (2)

TABLE 2

Estimated Number of Thermal Generating Plant Sites
500-Mw Capacity and Above for Year 1990

	Fossil-Fueled Plants by Mw Capacities					Nuclear-Fueled Plants by Mw Capacities				
	500 to 1,000	1,000 to 2,000	2,000 to 4,000	Over 4,000	Total	500 to 1,000	1,000 to 2,000	2,000 to 4,000	Over 4,000	Total
Northeast:										
Total Sites	22	15	4	--	41	7	19	17	2	45
New Sites	--	1	4	--	5	6	18	14	--	38
Cooling Towers	3	5	--	--	8	2	4	3	--	9
Southeast:										
Total Sites	15	12	7	--	34	10	22	21	7	60
New Sites	3	3	--	--	6	8	18	14	5	45
Cooling Towers	--	1	3	--	4	7	14	7	4	32
East Central:										
Total Sites	30	26	5	1	62	1	10	8	2	21
New Sites	5	9	1	1	16	--	8	7	2	17
Cooling Towers	7	8	1	1	17	--	4	1	1	6
South Central:										
Total Sites	37	31	23	1	92	3	9	9	1	22
New Sites	17	12	19	1	49	3	9	9	1	22
Cooling Towers	17	7	3	--	27	1	2	1	1	5
West Central:										
Total Sites	11	12	2	--	25	3	6	9	1	19
New Sites	1	5	--	--	6	2	4	5	--	11
Cooling Towers	1	4	1	--	6	1	4	3	--	8
West:										
Total Sites	14	19	4	1	38	4	7	9	13	33
New Sites	4	4	1	--	9	4	6	9	12	31
Cooling Towers	10	10	1	--	21	4	5	4	2	15
Total U.S.:										
Total Sites	129	115	45	3	292	28	73	73	26	200
New Sites	30	34	25	2	91	23	63	58	20	164
Cooling Towers	38	35	9	1	83	15	33	19	8	75

Water Requirements

Almost one-half of all water utilized in the United States is used for industrial cooling, including cooling water for condensing steam in power plants (3). Of the estimated 50 trillion gallons of water used by industry in 1964, approximately 81 percent was for electrical power production.

Depending upon the turbine cycle heat rate and the temperature rise of the circulating water selected in the plant design, approximately 300 to 900 gpm per mw must be pumped through the condensing system of a steam-electric generating plant.

The heat of condensation of the turbine exhaust steam is transferred to the circulating water and ultimately to the atmosphere through one of several methods.

Presently Used Methods of Rejecting Heat from Generating Stations

Once-through circulating water systems. Where sufficient volume of circulating water is available, as on a large river such as the Ohio or Missouri, circulating water is often taken directly from the river by means of an intake system, pumped through the condensers and then discharged back to the river at a location selected to prevent recirculation of the heated water back to the intake.

Generally, the once-through circulating system is the least expensive of the several types used, and utilities have used this method of providing circulating water wherever conditions permit.

Once-through systems are also used on natural lakes, ocean estuaries, rivers, including tidal rivers, flood-control reservoirs, water-storage reservoirs, navigational reservoirs and hydroelectric lakes.

Cooling lakes. Another method of providing circulating water for the steam-electric generating plant is to construct a pond or lake for the specific purpose of providing a source of circulating water and for dissipating the heat of condensation from the surface of the water. Generally, the surface area of the lake is sized for approximately one acre per mw of generating capacity, with variations above and below this figure in specific instances.

Wet-type cooling towers. Where insufficient water is available for a once-through circulating water system, an evaporative (wet-type) cooling tower is often used to dissipate the heat of condensation which has been transferred to the water in the condenser. Until the present concern about thermal pollution became prevalent, the use of evaporative cooling towers with steam-electric generating plants was generally with smaller plants and, primarily, in locations where insufficient water was available.

In the evaporative-type cooling tower, the circulating water is pumped through the surface condenser where it picks up the heat of condensation of the exhaust steam and is then pumped through the cooling tower where it is broken up into small drops by splashing down through the "packing" or "fill" of the tower.

More than 75 percent of the heat is removed by evaporation and the remainder is transferred to air by convection. After passing down through the tower fill, the water falls into a storage basin beneath the tower and is recirculated through the condenser.

Evaporative-type cooling towers are generally either of the crossflow design in which the air flows horizontally through the falling water or the counterflow design in which the air flows upward through the water.

Although the majority of cooling towers constructed in the United States have been of the mechanical-draft type which use motor-driven fans to move the air through the tower, natural-draft type towers are also constructed which utilize the stack effect of the tall tower structure for the movement of air. Approximately 20 natural-draft towers are either in operation or under construction in the United States (3). The natural-draft towers, usually hyperbolic in shape, are from 300 to 500 feet in height and over 300 feet in diameter at the base, with a height-to-diameter ratio of more than one but less than 1.5. Natural-draft towers are commonly used in Europe for steam-electric generating plants, where economics favor their selection over the mechanical-draft type.

During 1965 in the North Atlantic region as defined by the Federal Water Quality Administration (FWQA), only one plant (100 mw) out of 101 utilized cooling towers; in the Southeast region, two out of 61; in the Great Lakes region, none out of 54; and for the country as a whole, 116 out of 514 plants utilized cooling towers (Federal Water Quality Administration, 1968).

At the present time, approximately 30 states have adopted legislation concerning water temperature standards which have been approved by the Division of Standards of the FWQA (4).

The standards recommended by the National Technical Advisory Committee on Water Quality Criteria and the passage of state legislation controlling temperature rise can be expected to result in a reduction in the percentage of generating plants constructed for use with the once-through circulating water system and an increase in the use of evaporative water towers, where make-up water for such towers is available, and in dry-type cooling towers where make-up water is scarce or expensive.

Spray ponds, or spray canals. One method of rejecting heat from the condenser is to use a spray pond in conjunction with a recirculating supply of condensing water. In this method of rejecting heat to the atmosphere, the circulating water is sprayed into the air through discharge pipes and spray nozzles located above a pond which serves as a reservoir for receiving and holding the water for recirculation.

Except for certain small installations, spray ponds are not generally used for steam-electric generating stations.

Consumptive Use of Water by Generating Stations

All of the above methods of removing heat from the condensing exhaust steam of a turbine result in ultimate rejection of the heat to the atmosphere. Since a certain amount of cooling in each of these different methods is accomplished by the process of evaporating water, all of the methods result in a certain consumptive use of circulating water. In addition to losses by evaporation, the cooling towers, cooling ponds and spray ponds must waste a certain percentage of the water circulated in order to maintain the concentration of dissolved solids to limits compatible with operation without objectionable deposit of scale on the plant equipment.

The amount of evaporation experienced is dependent upon a number of variables, including:

1. Wet-bulb temperature of air.
2. Relative humidity.
3. Cloud cover.
4. Wind speed.
5. Range of cooling.

For a site in the northeastern United States, the evaporation-loss curve shown in Figure 2, assuming a relative humidity of 60 percent, cloud cover of 70 percent, wind speed at 8 mph, and a 20°F range with wet-bulb temperature varying from 40 to 80°F, is reproduced from (5). This curve shows the estimated water consumption, in gallons per kwh, for the following cooling methods:

1. Cooling pond with a surface area of 2 acres per mw.
2. Cooling pond with a surface area of 1 acre per mw.

3. Mechanical-draft evaporative cooling tower.

4. Spray pond.

5. Natural-draft evaporative cooling tower.

6. Natural lake or river.

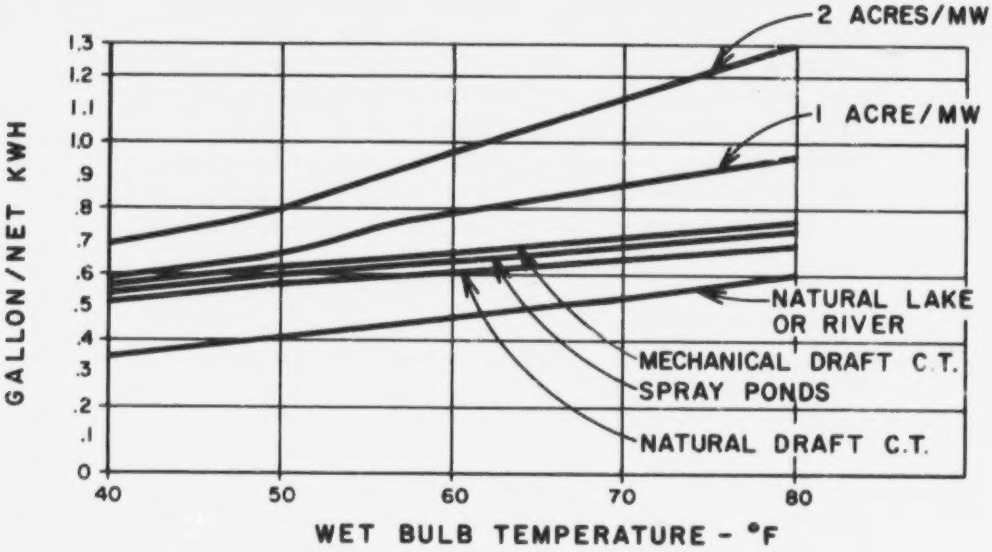


FIGURE 2— WATER CONSUMPTION VERSUS WET BULB TEMPERATURE (5)

At the present time, the water consumption from evaporation as a result of power generation is estimated to be 10 gallons per day per person and is expected to increase at a faster rate than the growth rate of power production because supplementary cooling systems, such as cooling towers and cooling lakes, will be utilized on a larger proportion of new generation capacity in the future.

Recent Legislation Governing Thermal Discharges to Natural Waters

Although there is much controversy as to the effects of temperature upon aquatic life, one fact is undisputed:

> Present and contemplated legislation sets definite temperature limits upon circulating water discharged from steam-electric generating plants.

For an excellent discussion of the effects of temperature on aquatic organisms, the reader is referred to (6).

The following is a brief review of existing and proposed laws governing environmental questions associated with the utility industry as reported in (7).

> A new law (H.R. 4148) would require the Atomic Energy Commission to obtain assurances that a nuclear plant will operate in conformity with applicable water quality standards. This is aimed primarily at the thermal pollution problem. Similar assurances would have to be obtained for other electric power plants which require federal permits or licenses such as the many power plants which require federal permits from the Corps of Engineers if their construction plans include structures on navigable waters.
>
> The National Environmental Policy Act passed in 1969 could have a bearing on federal activities in the power field, for it requires all federal agencies to consider environmental factors in carrying out their programs.
>
> The Federal Power Commission licenses only nonfederal hydroelectric plants and major electrical transmission lines and has authority to weigh the recreation, wilderness, fish and wildlife, and esthetic values of these projects.
>
> Only 20 states require licenses for new generating plants and most of them consider reliability and safety alone. However, there have been recent law enactments in some states for control over power plant siting. A 1968 Maryland law requires public hearings to consider the effects of the plant on the environment, including thermal effects. Washington, Vermont and Maine have recently passed similar legislation.
>
> The Federal Water Pollution Control Act, as amended by the Water Quality Control Act of 1965, authorizes the states and the Federal Government to establish water quality standards for interstate (including coastal) waters (8).

The water quality standards submitted by the states are subject to review by the Department of Interior and, if found to be consistent with Paragraph 3 of Section 10 of the Act, will be approved as Federal Standards by the Secretary of the Interior.

Paragraph 3, Section 10 reads as follows:

"Standards of quality established pursuant to this subsection shall be such as to protect the public health or welfare, enhance the quality of water and serve the purposes of this Act. In establishing such standards, the Secretary, the Hearing Board, or the appropriate state authority shall take into consideration their use and value for public water supplies, propagation of fish and wildlife, recreational purposes, and agricultural, industrial, and other legitimate uses."

If a state does not adopt water quality standards consistent with the above paragraph, the Act provides the Secretary with the opportunity to set the standards. In April, 1970, as reported in (4), 20 states had not yet received full approval of their water-temperature standards.

List of Generating Plants Equipped with Dry-Type Cooling Towers in Operation and Currently Under Construction

Table 3 shows a listing of the major steam-electric generating plants, either in operation or currently under construction, which are equipped with dry-type cooling towers.

TABLE 3

Generating Plants with Dry-Type Cooling Towers

Location	Rating	Type of Dry Tower	Year Commissioned
Rugeley, England*	120 mw	Heller	1962
Ibbenbüren, Germany*	150 mw	Heller	1967
Wolfsburg, Germany*	3-50 mw	GEA Direct	1961-67

TABLE 3 (continued)

Location	Rating	Type of Dry Tower	Year Commissioned
Grootvlei, South Africa	200 mw	MAN/Birwelco (Indirect)	1971
Gyöngyös, Hungary*	2-100 mw	Heller	1969
	2-200 mw	Heller	Under Constr.
Razdan, USSR	3-220 mw	Heller	1970-72
Wyodak, Wyoming, USA*	22 mw	GEA Direct	1969
	3 mw	Direct	1962
Utrillas, Spain	160 mw	GEA Direct	1970
Quetta, West Pakistan	7.5 mw	Baldwin-Lima-Hamilton (Direct)	1964
Bavaria	40 mw	GEA Direct	1960
Windhok, South Africa	3-30 mw	GEA Direct	1971
Switzerland	4.3 mw	GEA Direct	1969
Luxemburg	13 mw	GEA Direct	1956
Rome, Italy	2-30 mw	GEA Direct	1957
Cologne, Germany	28 mw	GEA Direct	1958
Sindelfingen, Germany	11&15 mw	GEA Direct	1960-61
Worms, Germany	5 mw	GEA Direct	1962
Chile	3.6 mw	GEA Direct	1963
Ludwigshafen, Germany	38 mw	GEA Direct	1966
Eilenburg, Germany	5.3 mw	Heller	NA
Dunaujvarus, Hungary	16 mw	Heller	1961

---

*Visited during study.

Description of Dry-Type Cooling Towers

General. The use of air for condensing turbine exhaust steam is not a new concept since it is reported that condensation by air cooling has been used in small industrial power plants for over 50 years (9). However, the application of air condensing to relatively large generating units has been limited, since the use of air condensation will generally result in an increase in construction costs.

There are two basic types of air-cooled condensing systems—the indirect system and the direct system. The indirect system utilizes a direct-contact condenser at the turbine to condense the exhaust steam. Water from the condenser is pumped to the dry-type tower for cooling and recirculation to the spray jets in the condenser. The indirect system is often referred to as the Heller system since the concept of the use of the indirect system of condensation by air cooling for use with a steam turbine-generator was presented by Dr. László Heller at the World Power Conference in Vienna in 1956 (10). Dr. Heller, who is Head of the Department of Energetics of the Technical University of Budapest, Hungary and also serves as Director of the Hungarian engineering firm called Höterv (charged with the development of dry towers), along with his assistant, Dr. L. Forgó, developed and perfected much of the special equipment required for use with the air-condensing systems—notably the heat exchanger coil, the automatic controls and the direct-contact condenser. However, a strict interpretation of the use of the term "Heller system" would limit it to an indirect system using the Heller-Forgó coil, since at least one indirect system using other coil designs is under construction.

In the direct system, steam is condensed in the coils without the use of a direct-contact condenser or circulating water.

Conventional Evaporative-Type System

An understanding of the conventional evaporative (wet-type) tower cycle is useful in considering the two types of air-cooled condensing systems. Figure 3 shows the schematic arrangement of an evaporative-type cooling tower serving a condensing turbine.

Condensing water is circulated through the tubes of a surface condenser and carries away the heat of condensation of the turbine exhaust steam. The exhaust steam comes into contact with the exterior surfaces of the tubes, and condenses as it gives up heat to the water.

The warm circulating water is piped to the evaporative cooling tower where it flows over the packing or fill, which may be closely spaced strips of asbestos-cement or wood, to break up the circulating water into small drops through which air is pulled by the tower fan. By a combination of evaporation and convection,

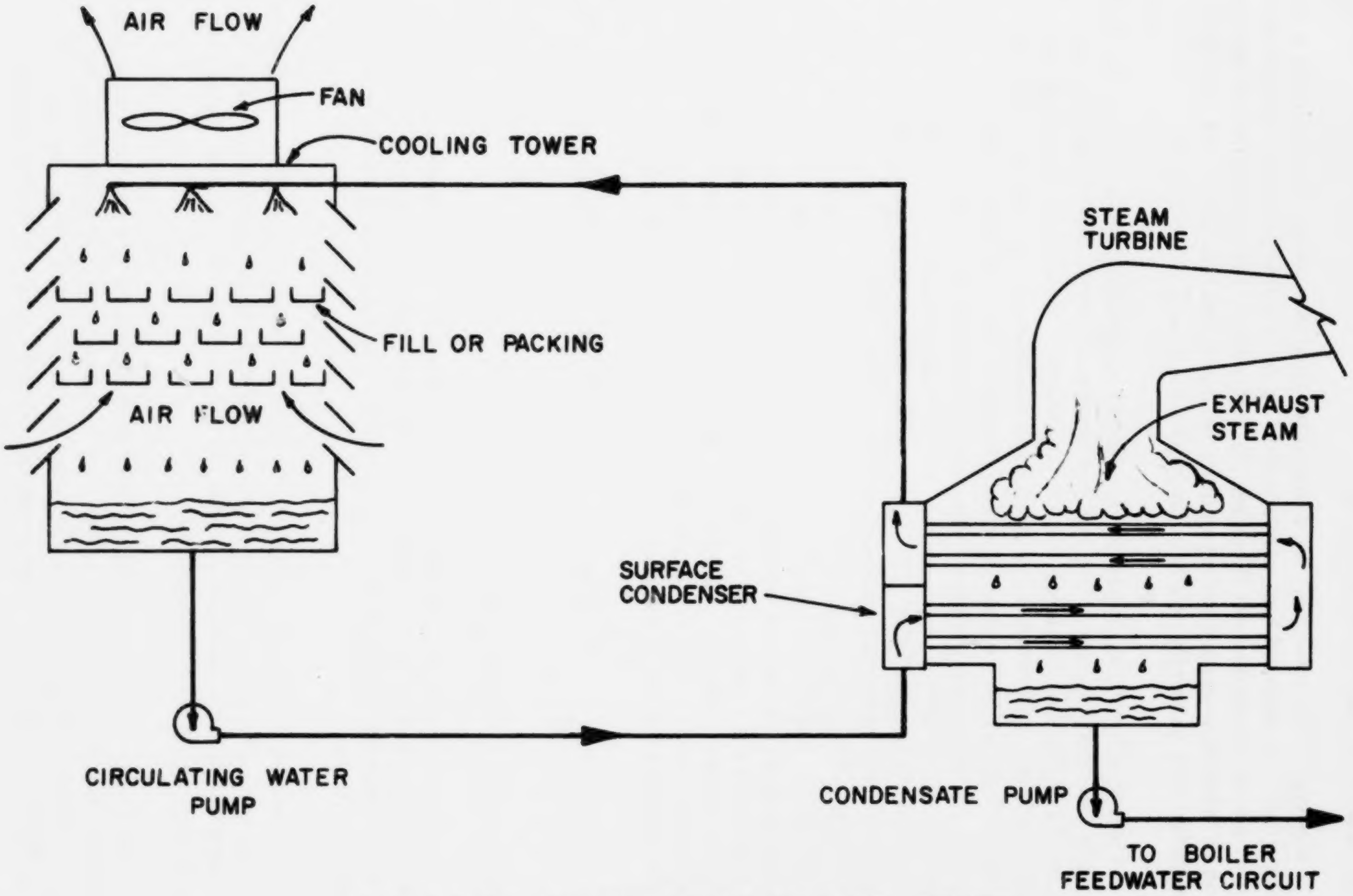


FIGURE 3 — EVAPORATIVE COOLING TOWER CONDENSING SYSTEM

the temperature of the circulating water is reduced and the water is again pumped through the condenser in a continuous cycle. The condensed steam (condensate) is removed from the condenser by the condensate pump and returned to the boiler feedwater circuit.

Dry-Type Systems

An explanation of the two basic air-cooled condensing systems follows:

Indirect system. For the indirect dry-type cooling tower, the principal components are:

1. A direct-contact steam condenser.
2. Circulating water pumps.
3. Water-recovery turbine (optional).
4. Cooling coils.
5. Means for moving air across the coils; either a natural-draft tower or a mechanical-draft fan.

Figure 4 shows a diagram of the indirect-type system with a natural-draft tower.

Either a mechanical-draft or a natural-draft tower is used with the indirect system. The choice is dependent upon the economics of each particular case, and such factors as fuel cost, comparative costs of construction of the two types, cost of money, and other pertinent factors are considered. Figure 5 shows the diagrammatic arrangement of an indirect dry-type cooling tower with a mechanical-draft tower.

Water from the cooling coils is sprayed into the direct-contact steam condenser and mixes directly with the exhaust steam from the turbine. The water from the tower and the condensed steam falls to the bottom where it is removed by circulating and condensate pumps. The greater part of the water flows through the pipes to the cooling coils, and an amount equal to the exhaust steam from the turbine is directed back to the boiler feedwater circuit for re-evaporation in the cycle. Since the cooling tower circulating water and the boiler feedwater are intimately mixed, the circulating water must be of condensate purity.

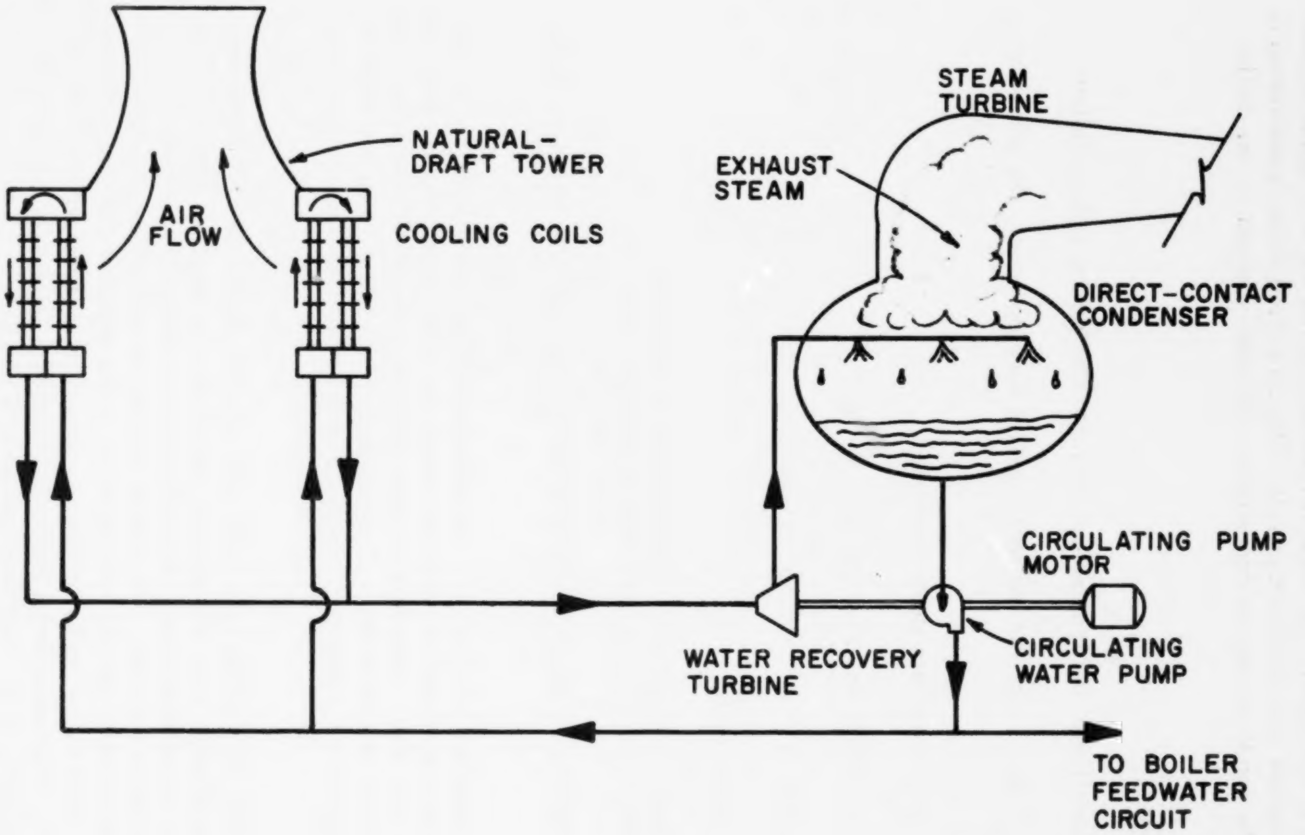


FIGURE 4 — INDIRECT, DRY-TYPE COOLING TOWER CONDENSING SYSTEM WITH NATURAL-DRAFT TOWER

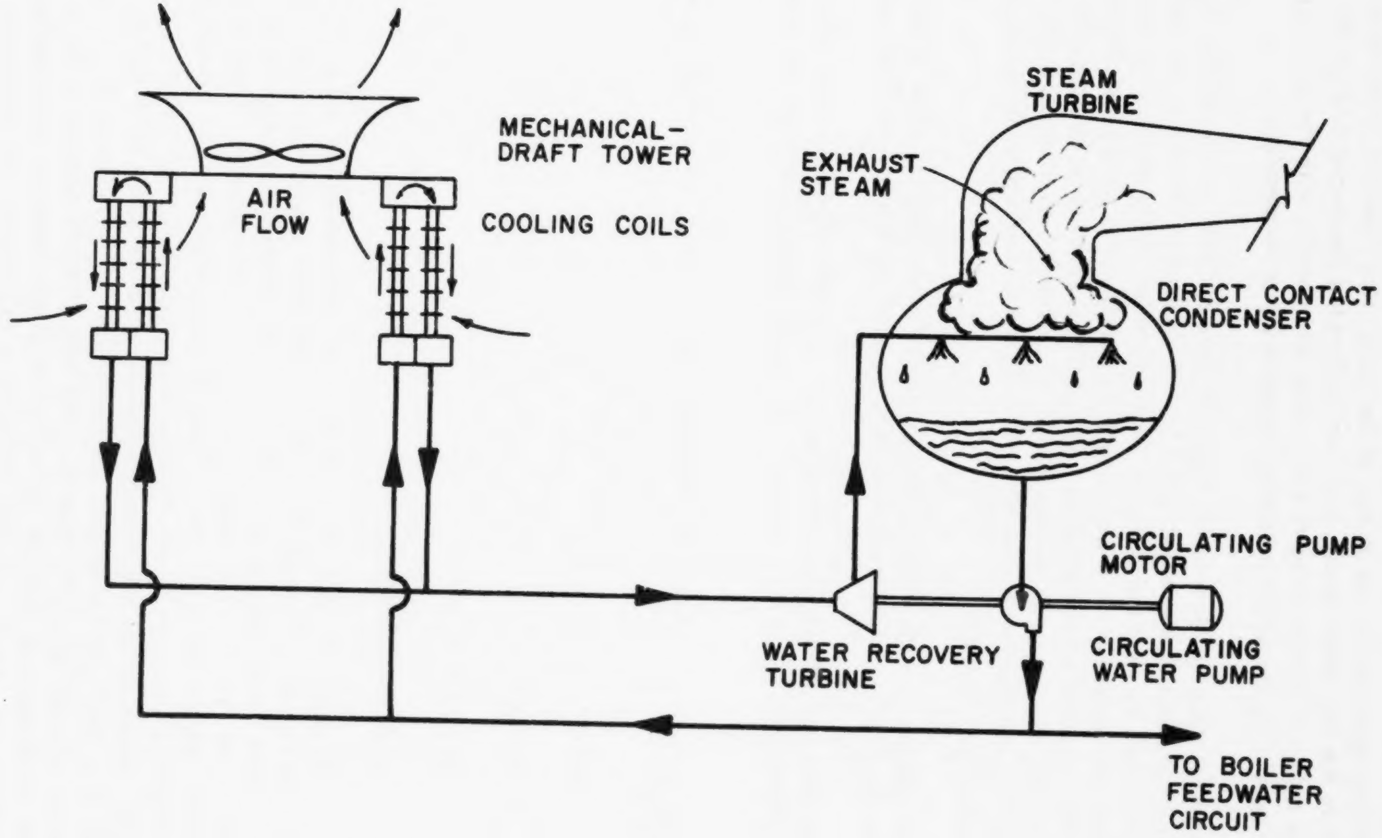


FIGURE 5 — INDIRECT, DRY-TYPE COOLING TOWER CONDENSING SYSTEM WITH MECHANICAL-DRAFT TOWER

In the Heller system, the cooling coils are mounted vertically, and the warm circulating water enters the bottom of the coils, flows upward in the inner rows of coils to the top water boxes, and then is directed downward through the outer rows of coils. The outer rows of coils come into contact with the entering air, thereby providing the greatest cooling range in water temperature.

To prevent drawing air into the system in case of leaks in the cooling coils, a positive pressure head of approximately 3 feet is imposed at the top of the coils. This is accomplished by means of either a throttling valve in the circulating water discharge from the tower, or, if a water-recovery turbine is used, by varying the position of the adjustable turbine vanes. In order to recover some of the pressure head between the cooling coils and the condenser, in some installations water-recovery turbines are coupled to the drive shaft of the circulating water pump to recover the available energy.

After passing through the recovery turbine, the circulating water is again sprayed into the direct-contact condenser and recycled through the cooling system.

Note that the circulating water does not come into direct contact with the cooling air; therefore, there is no evaporation loss of water as with the wet-type tower.

Direct system. The principal components of the direct air-cooled condensing system are:

1. Exhaust steam trunk.
2. Cooling coils.
3. Motor-driven fans.
4. Condensate pumps.

Figure 6 shows a diagram of a typical direct air-condensing system. Turbine exhaust steam is conveyed through the exhaust steam trunk, which is large in diameter to minimize the pressure drop, to the air-cooled coils where cooling air passing over the finned-coil surfaces condenses the steam. Shown here in the simplest form, the steam enters the top of the coil section and condenses as it travels downward with the steam and condensate flowing in the same direction. In actual installations, provisions are made for removal of noncondensable gases and air and for prevention of freezing during cold weather. The most common system in the United States is to use horizontal tube bundles with 80 to 90 percent of the tubes as the main condenser and 10 to 20 percent as an after-condenser to condense the remaining steam that is not condensed in the main condenser. The steam and con-

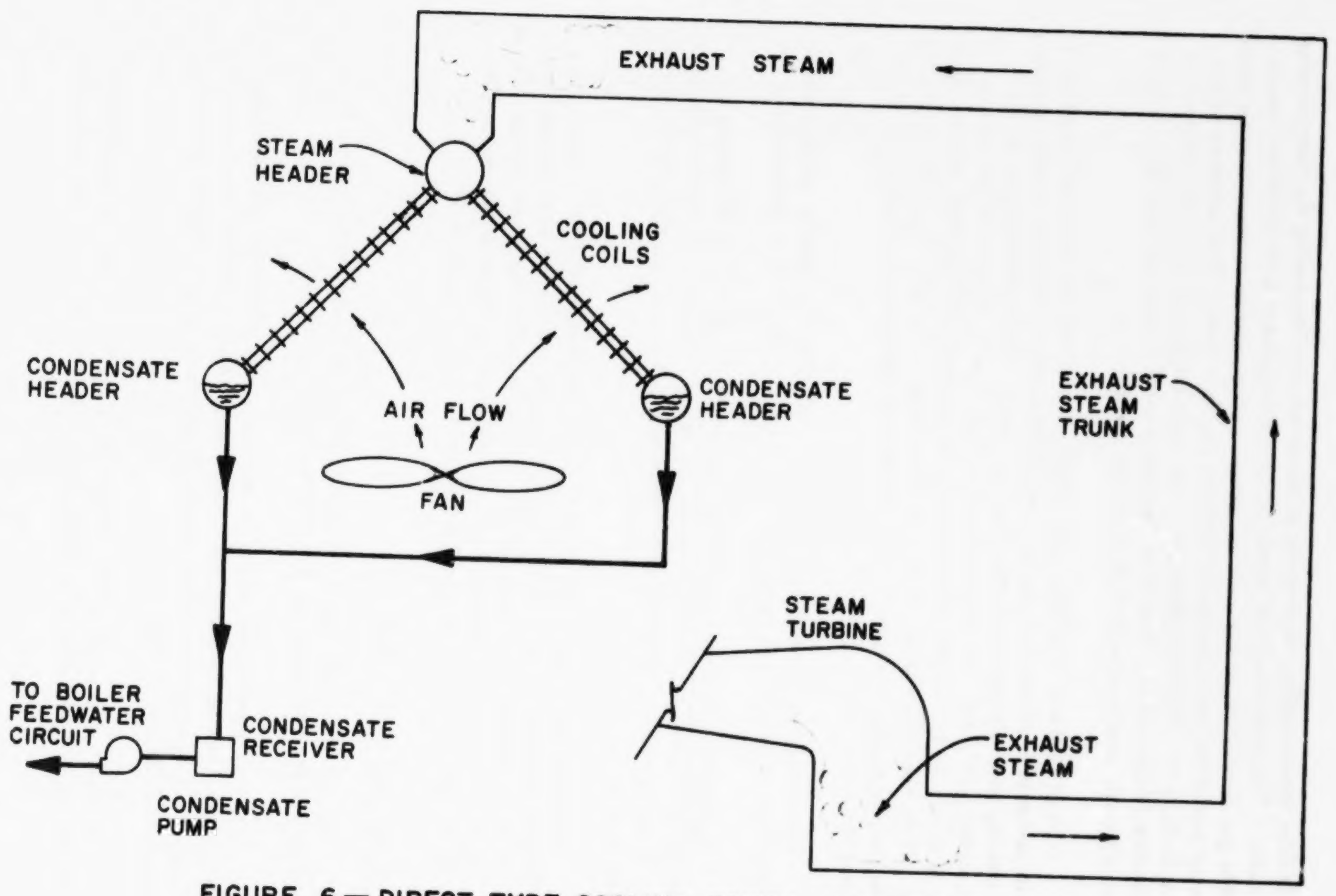


FIGURE 6 — DIRECT-TYPE COOLING TOWER CONDENSING SYSTEM

densate flow concurrently, minimizing pressure loss and increasing the heat transfer coefficient. The purpose of the foregoing coil arrangement is to minimize noncondensable gas blanketing of the main condenser as the residual noncondensable gases are swept out of the main condenser with the residual steam. The presence of excessive buildup of noncondensable gas in the main condenser would be deleterious to effective condensation. Freeze protection is usually accomplished by recirculation of warm air combined with the use of fan control.

GEA of Bochum, Germany uses a method of direct condensation in which a certain percentage of the cooling coils are constructed so that the remaining steam, after passing down through a condensing unit, enters the bottom headers of the aftercooling coils, and the condensate and steam flow in opposite directions in order to obtain better control of condensate temperature during cold-weather operation. Only noncondensables remain in these latter coils near the upper ends after all the steam has been condensed, thus preventing freeze-up in that region of the heat exchangers.

The condensed steam from the cooling coils flows by gravity to condensate receivers from which it is pumped back to the boiler circuit by a condensate pump.

Comparison of indirect and direct systems. The principal difference between the two systems is the large volume of exhaust steam which must be handled in the direct system as compared to the smaller volume of circulating water in the indirect system.

Although discussions with users of the direct systems did not indicate that any adverse experiences as a result of condenser air leaks have been encountered with the direct systems, the fact that all the cooling coils are under a high vacuum during operation is sometimes considered a disadvantage when compared to the indirect systems with positive water pressures in the cooling coils (9).

Use of Air Cooling by Industry

The use of finned-tube heat exchangers to dissipate waste heat to the atmosphere has been accepted by industry for well over 75 years.

Common applications of the finned-tube heat exchanger are the automobile radiator and steam or hot-water heating systems. Also, radiator-type heat exchangers have been used on stationary, internal-combustion, engine-driven generators up to 3,000 kw in size.

In recent years, especially since the late 1940's, industry has turned more to the use of air cooling for discharging large amounts of heat to the atmosphere in processes where heretofore water-type heat exchangers or evaporative-type cooling towers have been used.

In 1969, it was estimated that the chemical process industry spent $50 million on air coolers (11).

The experience gained by the chemical and process industries in air cooling can be of value to the electrical generating utilities as they consider air-cooled condensing systems.

Extent of air cooling in industry (13). Industrial air cooling in natural-gas processes was first used approximately 30 years ago in the arid south-central and southwestern parts of the United States where water is not readily available. Later, the petroleum refining industry, petro-chemical and chemical process industries began to use air cooling on a wide scale.

Plants originally equipped with water-cooled systems and new process plants often utilize air to remove from 60 to 100 percent of the waste heat. Process plants with indirect air-cooling systems as large as 2 billion Btu per hour—the heat rejection equivalent of a fossil-fueled electrical generating plant of over 400 mw—have been built by the Hudson Products Corporation of Houston, Texas. Direct condensing systems have been supplied, on a smaller scale, to these same process industries.

The selection of air cooling is being done on the basis of economic justification, taking into account first cost, operating and maintenance costs, and plant capability. Important factors in the evaluation of air cooling are the increasing costs of securing, treating and disposing of cooling water, and environmental limitations.

The problems peculiar to the various processes have resulted in the use of many exotic materials for the air-cooled coils, including carbon steel, alloy steel, stainless steel, nickel, copper admiralty, aluminum, cupro-nickel, hastelloy, titanium and karbate. Process coil design pressures and temperatures often far exceed those required for steam-electric plant air-condensing systems, ranging up to 15,000 psi and 1,000°F.

The most common tube size is 1 inch O.D., although diameters from 5/8 inch to 1-1/2 inch are used, with tube lengths up to 40 feet. Except for special high-temperature process coolers, or for economizers where combustion products are in contact with the fins, fin material is predominately aluminum.

Use of Air Cooler with Refuse Incinerators

An important use of air-cooled heat exchangers in recent years in Europe has been that of steam condensing for municipal-type refuse incinerators. Modern refuse incineration makes use of a water-cooled furnace in place of the refractory-lined furnaces previously used. In order to eliminate odors and to insure that all putrescible material is consumed, the furnace temperature of the incinerator should be in the range of 1,500°F to 1,850°F. These high temperatures, when used with the older type incinerators, resulted in high maintenance costs for the refractory-lined furnaces; whereas, water-cooled furnaces can withstand high furnace temperatures without deterioration.

In order to absorb the heat from the combustion, steam is generated in the process. In certain instances where there is a ready market for the steam, it is used for heating, process or power generation. However, there are installations where it is not feasible or practical to sell the steam generated in the incineration process and, in order to condense the steam and reuse the condensate, air-cooled condensers have been used. A number of incinerators have been constructed where all steam is condensed in the air condenser and the condensate recycled through the boiler. There are others where the air-cooled condenser is used during seasons where there is no market for the steam.

Refuse incinerators which utilize air-cooled condensing systems have been installed at the plants listed in Table 4.

TABLE 4

Refuse Incinerators with Air-Cooled Condensing Systems

Location	Condensing Capacity (lbs. of steam/hr.)	Condensing Pressure	Construction Year
Darmstadt, Germany	67,000	610 psi	1967
Vienna, Austria	97,000	280 psi	1970
Biel, Switzerland	18,250	298 psi	1967
Bremen, Germany	220,000	212 psi	1968
	66,000	18.5 psi	1968

TABLE 4 (continued)

Location	Condensing Capacity (lbs. of steam/hr.)	Condensing Pressure	Construction Year
Hagen, Germany	35,000	214 psi	1966
	35,000	214 psi	1966
Bad Godesberg, Germany	22,000	156 psi	1966
Toulouse, France	45,000	44 psi	1969

From: GEA Air-Cooled Steam Condenser Summary of Installations.

SECTION II

FUNDAMENTALS

Design and Construction Considerations

The trend in industrial air-cooling design is to standardization of basic unit components including the air-handling unit, supports and structures, tube bundles, and rotating mechanical parts, with maximum shop assembly and testing.

V-belts as well as direct-coupled, in-line, gear-reducer motors are used to drive the fans. Fans with a minimum of 4 to 8 blades with blade widths up to 18 inches are used. Fans with diameters of 8 to 14 feet are common in industrial air coolers.

Figure 7 shows a cross section of a typical industrial air-cooled heat exchanger.

Codes and Testing (13)

Except for proprietary header designs based on actual strain-gauge tests, Section VIII for Unfired Pressure Vessels of the ASME Code is generally used for header construction.

Field test codes are in the process of being developed for air-cooled equipment by committees of the ASME and AICHE, and there is currently an American Petroleum Institute (API) specification.

Fan ratings are ordinarily based on wind-tunnel tests in accordance with Bulletin 210, April 1962 Edition of Standard Test Code for Air Moving Services, adopted by the Air Moving and Conditioning Association. Noise tests are conducted in accordance with the United States Acoustical Society standards with current maximum limitations as required by the Walsh-Healy Act, effective July, 1969. The trend is towards even more stringent noise limitations which usually results in an increase in first cost and in operating costs.

Fin Types

General. In general, there are five types of fins used with industrial air coolers. Figure 8 shows a cross-section view of these and also the triangular pitch used with circular tubes. The fin tube cross sections are taken through Section AA of Figure 8(a).

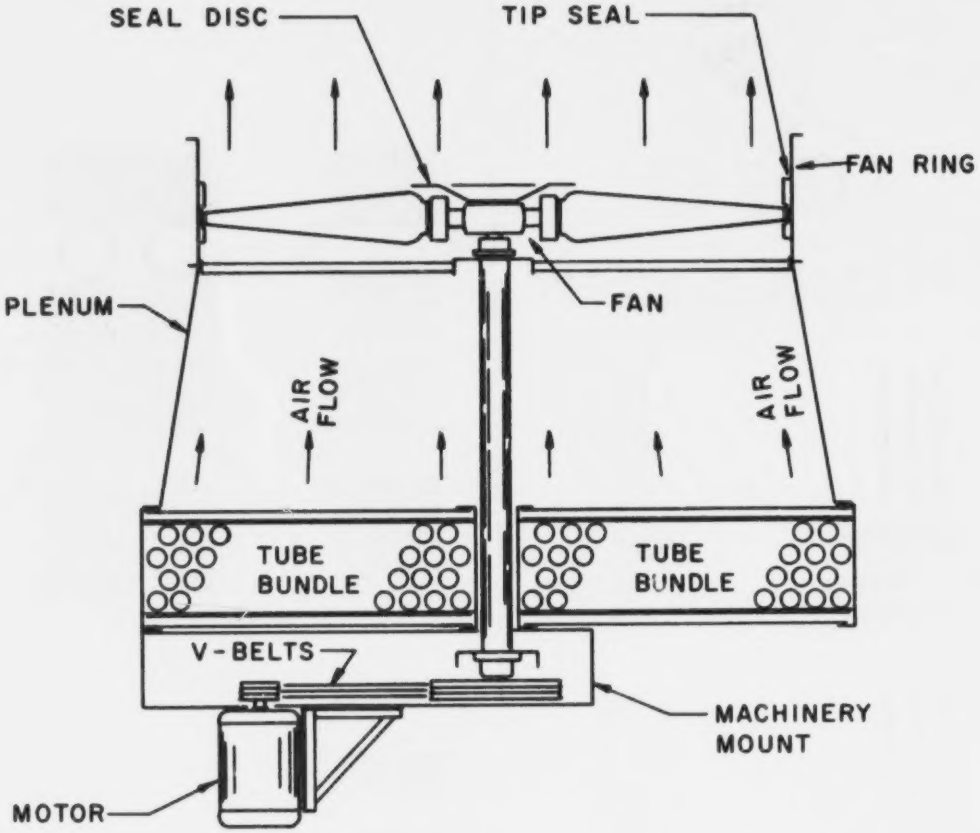
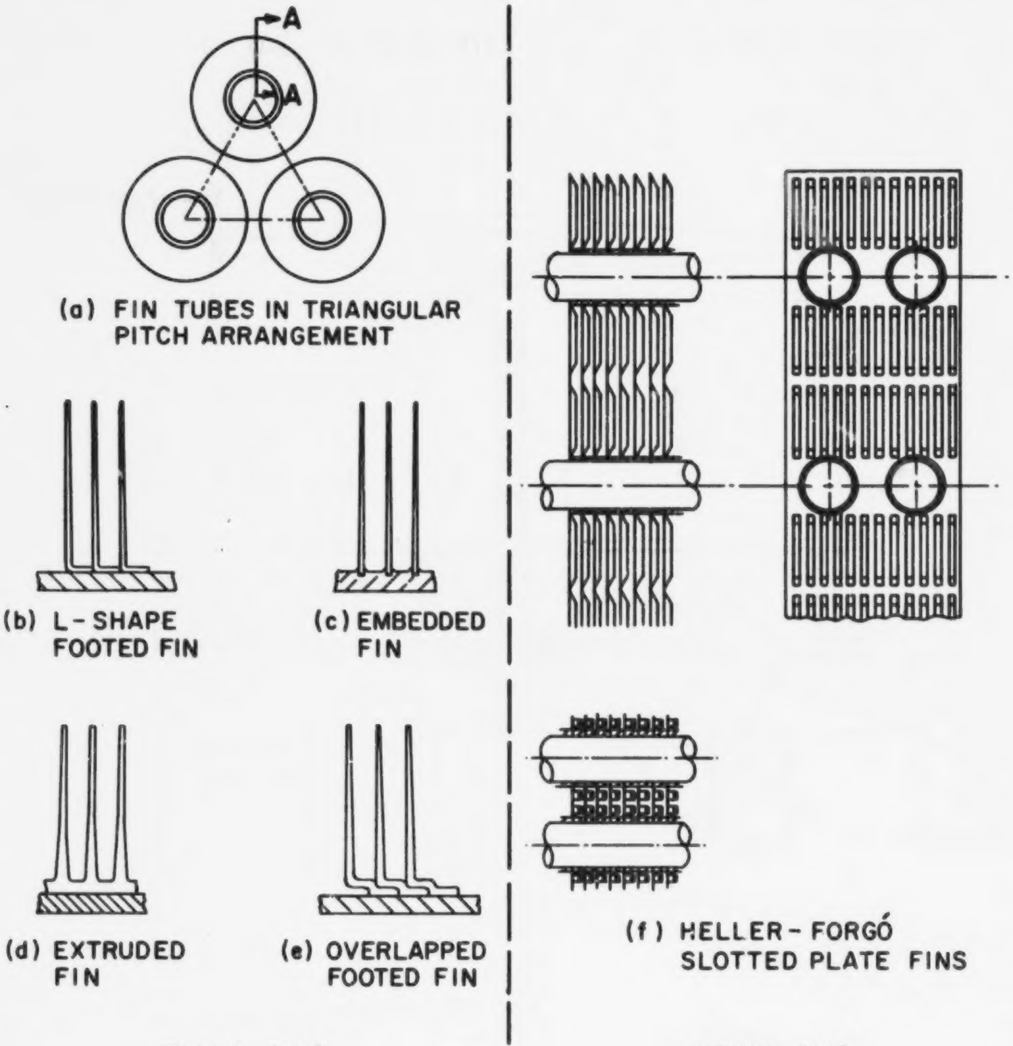


FIGURE 7 — HORIZONTAL AIR COOLED HEAT EXCHANGER (12)



(a) FIN TUBES IN TRIANGULAR PITCH ARRANGEMENT

(b) L-SHAPE FOOTED FIN

(c) EMBEDDED FIN

(d) EXTRUDED FIN

(e) OVERLAPPED FOOTED FIN

(f) HELLER-FORGÓ SLOTTED PLATE FINS

FROM (14)

FROM (15)

FIGURE 8—HEAT EXCHANGER—FIN TUBE TYPES

Tension-wound, footed, Figure 8(b). This type of finned tube is generally used for temperatures up to approximately 225°F. The fins are predominately aluminum and the tubes of commercially available material suitable for the service. Specially designed machinery wraps the fin around the tube. The L-shaped foot provides heat transfer surface between the tube and the fin. This is the least expensive type of finned tube, but has the poorest bond and the least thermal capability.

Embedded fin, Figure 8(c). This fin type is used for temperatures up to 750°F. The fins, usually of aluminum or steel, are wrapped around the tube and fitted into grooves which have been cut or rolled into the tube. The embedded fin is locked into place in the wrapping process by rollers which press tube metal against the base of the fin to form the necessary bond. This fin tube is the most versatile and physically rugged tube.

Extruded fin, Figure 8(d). This fin is used for medium-temperature service up to 550°F. The finned section of the tube is extruded from a heavy-wall aluminum outer tube which has been fitted over the outside of an inside tube of suitable material for the particular service requirement. The extrusion process provides the necessary bond between the inner and outer tubes. This type of tube has been used for chemical process applications where corrosion has been a problem and in cases where supplementary water sprays are used.

United States manufacturers use this type of fin extensively for the process industries, and offer slotted plate-type fins as well as extruded fins for the utility industry.

Wrapped-on overlapped, footed fin, Figure 8(e). This fin is used for temperatures up to 450°F. The tube is constructed by wrapping the fin around the tube and is the same as the tension-wound, footed fin, except for the double foot which affords better protection against corrosion.

Plate-type fin, Figure 8(f). In addition to the above-described finned tubes, the plate-type fin is also used in industrial work. In the plate-type construction, flat plates are drilled or punched for the tube and the plates are placed over the tubes. In the United States, plate-type finned coils are always furnished with collars integral with the fins, but in Europe separate collars are often used. The purpose of the collar is to increase the contact surface between the tube and the plate fin. As shown in Figure 8(f), the Forgó coil, developed for use with the Heller system, is of the plate-fin design. Note the slots or louvers in the aluminum fins. The purpose of the slots is to improve heat transfer rate by preventing the thickening of the boundary layer which forms with uninterrupted air flow over a flat surface. Since each slot results in another boundary layer, the heat transfer is improved.

Types of Air-Cooled Exchange Systems

There are a number of different types of systems used in air-cooling practice in industry. Figure 9, from (14), illustrates the most commonly used systems.

Direct air cooling, as illustrated in Figure 9(a), is commonly used and is the simplest system.

Direct air cooling with warm air recirculation, Figure 9(b), is used to prevent freezing of process fluid during cold weather, or where it is desired to keep the cooling air temperature high. Air coolers have been used where air temperatures reach 50°F below zero with this design.

In the indirect cooling system with a liquid-cooled exchanger, Figure 9(c), the process fluid is cooled in a shell-and-tube heat exchanger by a cooling fluid which is recirculated in the secondary air-cooled heat exchanger.

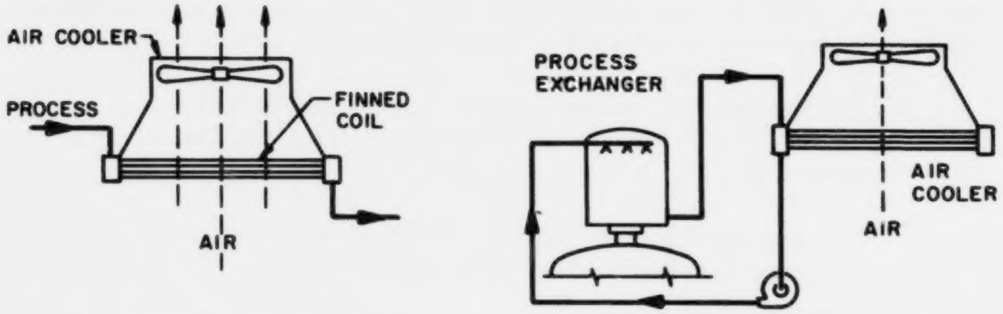
The indirect cooling system with spray condenser, Figure 9(d), is similar to the indirect cooling system with a liquid-cooled heat exchanger, Figure 9(c), with the substitution of a spray condenser for the closed heat exchanger, and closely resembles the Heller indirect air-cooling system for steam-electric generating plants.

The combination air cooler-cooling tower unit, Figure 9(e), performs a double function of cooling air by humidification spray to supply the air cooler with inlet air of lower temperature than the available ambient dry-bulb temperature and also supplies cooling water which is used to remove process heat from miscellaenous cooling services.

The direct, V-type, air-cooled exchanger with sprays, Figure 9(f), requires less space than the conventional horizontal air cooler, but is more expensive for a given duty and requires more horsepower. By spraying water of condensate or demineralized quality onto the fin tubes during a few hours of extremely high dry-bulb air temperature, the process outlet temperatures from the cooling coils can be reduced.

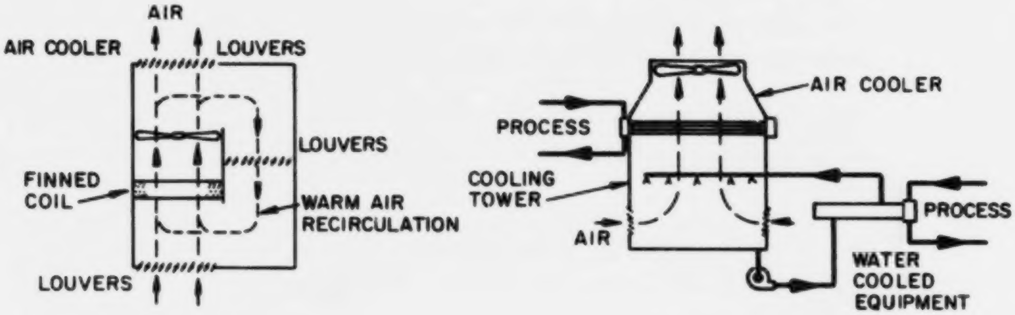
Theory of Heat Transfer from Air-Cooled Coils

Although it is recognized that there are differences of opinions among designers of air-cooled heat exchangers as to methods of determining heat rejection performance and, consequently, the material used herein from various references is not always in agreement, it is felt that all pertinent information available should be included in order to present as complete a picture as possible of the state of the art of dry-type cooling towers.



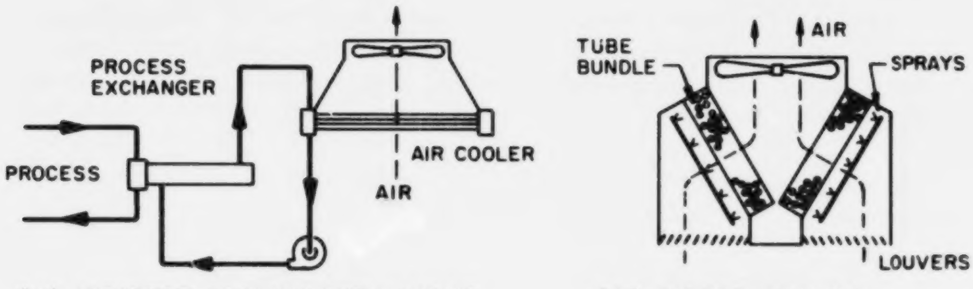
(a) DIRECT COOLING

(d) INDIRECT COOLING WITH SPRAY CONDENSER



(b) DIRECT COOLING WARM AIR RECIRCULATION

(e) COMBINATION AIR COOLER-COOLING TOWER UNIT



(c) INDIRECT COOLING WITH LIQUID COOLED EXCHANGER

(f) DIRECT COOLING-"V" TYPE AIR COOLED EXCHANGER WITH SPRAYS

FIGURE 9 — EVAPORATIVE-TYPE HEAT EXCHANGER SYSTEMS (14)

Basic theory (12). The resistance to the flow of heat from a hot fluid inside a finned tube to cooler air flowing across the outer surfaces of a clean, corrosion-free tube can be expressed as six separate components of resistance:

1. The internal film resistance to the flow of heat between the hot fluid in the tube and the internal surfaces of the metal tube.
2. The resistance to conduction of heat through a fouling resistance deposited on the inside wall of the tube.
3. The resistance to conduction of heat through the metal wall of the tube.
4. The resistance to flow of heat across the bond or gap between the inner tube metal and the fin muff or collar.
5. The resistance to flow of heat through the fin from the inner periphery of the fin to the outer periphery of the fin.
6. The air-film resistance to the flow of heat from the surface of the fin to the air passing over it.

Of the six resistances, the air-film resistance is the most significant. The other resistances are, in general, relatively low compared to the air-film resistance which impedes the flow of air from the fin surface to the air. Because of the greater resistance to transfer of heat from the metal fins to the air, it is necessary to increase the heat transfer surface in contact with the air, which accounts for the use of fins in air-cooled heat transfer surface. Typical ratios of fin surface to tube surface are from 10 to 30.

The transfer of heat from the inside fluid to the air is influenced by a number of variables:

1. The temperature difference between the fluid and the air.
2. The design and surface arrangement of the coil.
3. The velocity and character of air flow across the tubes.
4. The velocity and physical properties of the fluid inside the tubes.

The driving force for the transfer of heat between the fluid inside the tubes and the air flowing across the tubes is a function of the logarithmic mean temperature difference (LMTD) between the fluid and the air. The LMTD is expressed by the following formula:

$$\text{LMTD} = \frac{\text{GTTD} - \text{LTTD}}{\text{Log}_e \left[\frac{\text{GTTD}}{\text{LTTD}} \right]} \qquad [1]$$

where: LMTD = logarithmic mean temperature difference, °F

GTTD = greater terminal temperature difference between the hot fluid and the cold fluid, °F

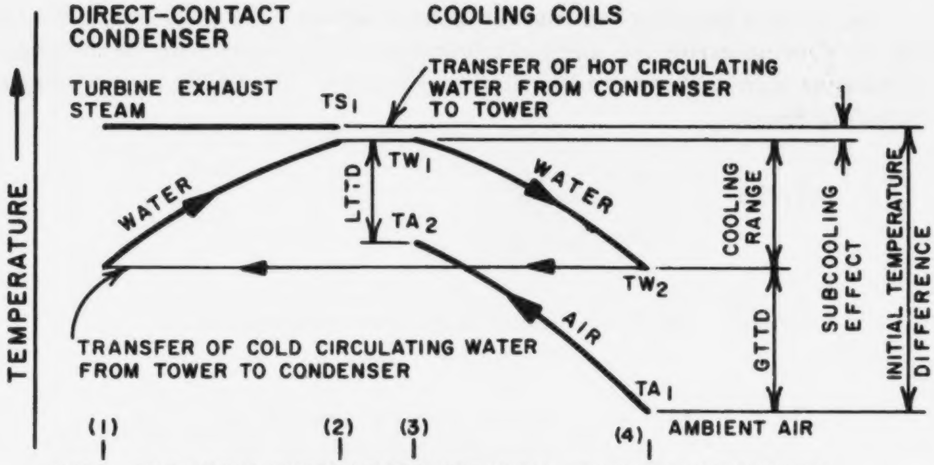
LTTD = lesser terminal temperature difference between the hot fluid and the cold fluid, °F

Figure 10 illustrates the basic temperature diagram as it applies to dry-type cooling tower coils.

Indirect system. Figure 10(a) shows the temperature relationship that exists in the indirect system. The left side of Figure 10(a) represents the temperatures that exist in the direct-contact condenser where the cool circulating water from the tower mixes with the turbine exhaust steam. The upper line represents the temperature level of the condensing steam. Since condensation takes place at a constant temperature corresponding to the saturated steam temperature of the turbine back pressure, this line is horizontal and at temperature T_{s1}. The lower curve on the left side of the diagram represents the temperature condition of the circulating water heated from T_{w2} to T_{w1} as the exhaust steam transfers the heat of condensation to the water.

The difference in temperature between T_{s1} and T_{w1} represents subcooling of the condensate and circulating water below the saturated steam temperature of the exhaust pressure and is a thermal loss to the turbine cycle. In the typical direct-contact condenser, the subcooling is approximately 3°F, but it is possible to have a condenser in which no subcooling exists, in which case T_{s1} and T_{w1} are the same.

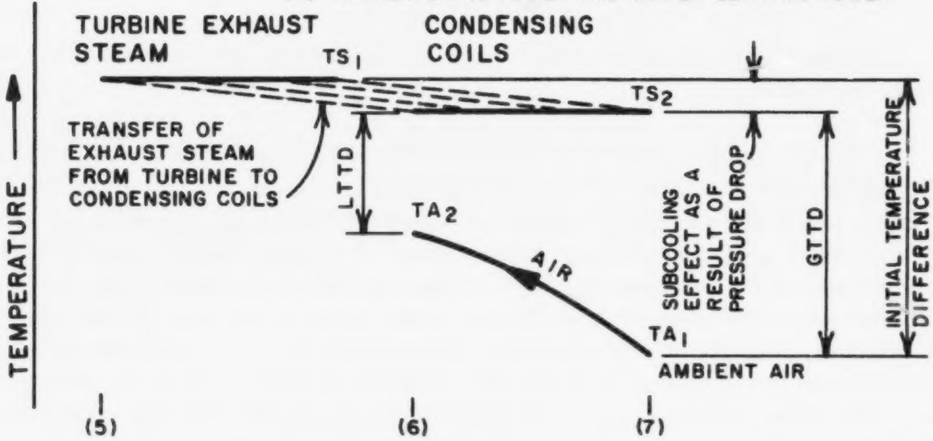
The diagram on the right-hand side of Figure 10(a) illustrates the temperature condition of the circulating water and the cooling air as the water flows through the coils. The air at ambient temperature T_{a1} comes into contact first



NOTE: THE ABOVE SKETCH DOES NOT IMPLY FLOW RELATIONSHIPS

(a) INDIRECT SYSTEM

(1) WATER AND STEAM ENTERING CONDENSER
(2) WATER LEAVING CONDENSER
(3) WATER ENTERING TOWER AND AIR LEAVING TOWER
(4) AIR ENTERING TOWER AND WATER LEAVING TOWER


TURBINE EXHAUST STEAM
TS1
CONDENSING COILS
TS2
TRANSFER OF EXHAUST STEAM FROM TURBINE TO CONDENSING COILS
LTTD
TA2
AIR
SUBCOOLING EFFECT AS A RESULT OF PRESSURE DROP
GTTD
INITIAL TEMPERATURE DIFFERENCE
TA1
TEMPERATURE
AMBIENT AIR
(5)
(6)
(7)

(b) DIRECT SYSTEM

(5) STEAM LEAVING TURBINE
(6) STEAM ENTERING CONDENSING COILS AND AIR LEAVING CONDENSING COILS
(7) AIR ENTERING CONDENSING COILS AND HOT WATER (CONDENSATE) LEAVING COILS

FIGURE 10 — TEMPERATURE DIAGRAMS OF DIRECT AND INDIRECT DRY COOLING TOWER HEAT-TRANSFER SYSTEMS

with the cooled water at T_{w2} and is heated to T_{a2} as the water cools from T_{w1} to T_{w2}. The diagrams shown are for counterflow of air and water. In actual practice, crossflow correction factor is used to compensate for the deviation in heat exchanger performance because of the crossflow condition.

Direct system. Figure 10(b) shows the temperature relationship between the turbine exhaust steam and the cooling air as they exist in the direct air-condensing system. No circulating water is used in the direct condensing system and the exhaust steam with temperature at T_{s1} is conveyed through the exhaust steam trunk to the condensing coils. The difference in temperature between T_{s1} and T_{s2} is the result of pressure drop in the exhaust steam trunk and also is a loss to the cycle; T_{s2} represents the temperature level in the coils at which condensation takes place and is at a constant level since the steam condensation takes place at a saturated temperature corresponding to the steam pressure in the coils. The lower line of this diagram shows the temperature rise of the air as it flows past the coils and picks up the heat of condensation.

Further subcooling below temperature T_{s2} can result from improper design or operation of the condenser.

The heat transfer from the coils to the air is expressed by the general formula:

$$Q = U \, \text{LMTD} \, A \, F_g \qquad [2]$$

where: Q = total heat transfer of the coil, Btu/hr.

U = over-all coefficient of heat transfer, Btu/(hr. ft.^{2} °F)

LMTD = logarithmic mean temperature difference between fluid inside the coil and the air, °F

F_g = dimensionless crossflow correction factor — usually around 1.0

A = area of the coil surface, ft.^{2}

The over-all heat transfer coefficient (12) must be applied to the proper area. It is sometimes applied to the inside surface of the tubes, sometimes to the outside surface of the bare tubes and sometimes to the total outside extended surface. In any case, the area chosen must be consistent with the properly applied over-all U.

The over-all resistance to heat flow is the sum of the six individual resistances, which are set forth on page 30 of this report.

Since the air-film resistance is a function of the velocity of the air and the geometry of the fin surface only, and since the efficiency of the fin is a function of the air-film coefficient, the geometry of the fin surface, and the conductivity of the metal of the fin, these resistances are usually combined into one resistance for a particular geometry and fin metal. This one resistance is then only a function of air velocity and is usually determined from wind-tunnel tests for any particular surface.

Thus, the over-all U and R can be expressed by the following equations:

$$R = r_a + \frac{A_o}{A_i}\left[r_g + r_f + r_m + r_i \right] \qquad [3]$$

or

$$\frac{1}{U} = \frac{1}{h_a} + \frac{A_o}{A_i}\left[r_g + r_f + \frac{t_t}{k_t} + \frac{1}{h_i} \right] \qquad [4]$$

where: A_o = area of the extended surface, ft.^{2}

A_i = area of the inside tube surface, ft.^{2}

h_a = apparent coefficient of heat transfer of a finned surface, Btu/(hr. ft.^{2} °F)

h_i = coefficient of heat transfer on inside of tube, Btu/(hr. ft.^{2} °F)

r_a = resistance to air (°F hr.)/Btu

r_i = resistance to flow from fin surface to air (°F hr.)/Btu

r_m = resistance to flow through metal (°F hr.)/Btu

R = thermal resistance (°F hr.)/Btu

t_t = tube thickness, ft.

k_t = thermal conductivity of the tube metal, Btu/(hr. ft.^{2} °F)

r_g = gap or bond resistance (°F hr.)/Btu

r_f = fouling resistance (°F hr.)/Btu

The apparent coefficient of heat transfer of the external surface, h_a, is, as noted above, a combination of the heat transfer from the collar of the fin which has 100 percent efficiency and the heat transfer from the fin which has incorporated in it the average resistance to flow of heat through the fin metal to every part of the fin surface. The apparent coefficient of heat transfer of the external surface can be expressed as follows:

$$h_a = h_f \left[1 - (1 - \eta_f) \frac{A_f}{A_o} \right] \qquad [5]$$

where: A_f = surface area of fins, ft.^{2}

A_o = total area of fin surface and collar, ft.^{2}

h_f = mean surface coefficient of heat transfer of a finned surface, Btu/(hr. ft.^{2} °F)

η_f = fin efficiency

Since the ratio A_f/A_o is usually .91 to .97, this is not a significant correction. Fin efficiency can be calculated by methods of Gardner (16) or others, and correlation exists for approximate calculations of h_f for many geometries [see Kays and London (17)]. However, the only truly accurate method of determining h_a for any particular geometry is by wind-tunnel tests where h_a is plotted against face velocity of the air.

Since fin efficiency is a function of fin height and fin thickness, and since surface ratio is also a function if fin height, the design of a fin surface is an economic balance between increasing fin height, and consequently surface ratio, and decreasing fin efficiency.

Mechanical fabrication technology also imposes limits on increasing surface ratios.

The fin efficiency, η_f, is given by the following formula:

$$\eta_f = \frac{\tanh\left(r_t \varphi \sqrt{\frac{2h_f}{k_f t_f}}\right)}{r_t \varphi \sqrt{\frac{2h_f}{k_f t_f}}} \qquad [6]$$

where: $\varphi = g\left(\frac{r_f}{r_t}\right)$

t_f = fin thickness, ft.

r_f = radius of curvature of fin tip, ft.

r_t = radius of curvature of fin root, ft.

h_f = mean surface coefficient of heat transfer of a finned surface, Btu/(hr. ft.^{2} °F)

k_f = thermal conductivity of fin, Btu/(hr. ft.^{2} °F)

g = denotes "a function of"

Good design of a fin surface dictates obtaining the maximum air side film coefficient for a minimum expenditure of pressure loss of air passing through the surface. This can be accomplished by louvering or serrating the fin, thus interrupting the boundary layer of air which is the resistance to heat transfer. This interruption of the boundary layer considerably increases the air-film coefficient at a modest increase in pressure loss.

Good heat transfer from finned coils is also dependent upon a good mechanical bond between the tube and the fin. The effectiveness of the bond depends on the as-manufactured compression pressure between the fin collar and the tube. For low compression pressures it is possible for only a fraction of the two surfaces to be in contact. This results in an air gap over part of the contacting surfaces and consequent loss of heat transfer. As the metal temperature of the coil rises from the as-manufactured temperature, if the fin is aluminum and the tube is steel, the aluminum will expand more than the steel, thereby relaxing the initial contact pressure.

For finned coils, the heat transfer to the air stream is dependent upon so many factors that reliable rating and performance information for any specific coil design must be verified by actual tests (18).

Design of air coolers (12). The design of an air cooler for any process condition involves several trial-and-error procedures. The size of the air cooler is not initially known and, therefore, the exit air temperature will not be known and neither will the transfer rate in the tubes, since velocity in the tubes cannot be calculated.

The initial step is to assume an air temperature rise in the air cooler compatible with the process fluid inlet and outlet temperatures and to calculate the LMTD based on this assumption. A "U" is assumed, based on experience with similar equipment. From the calculated LMTD and assumed "U", a required surface can be calculated. A trial arrangement of this surface is made in the most economical way as to length of tube, width and number of bundles, and depth of tube rows. With this arrangement and assuming an air face velocity compatible with desired pressure loss, the temperature rise of the air is calculated. If this rise does not match the assumed rise, the cooler must be rearranged until it does match by varying the surface, or face velocity of the air, or both. Then the tube side passes can be arranged to suit the required pressure drop, and the tube side film coefficient can be calculated. If summation of the individual resistances does not equal the assumed "U", the process must be repeated until balance is achieved. Knowing now the number of crossflow passes, the LMTD correction factor must be incorporated.

Initial temperature difference. Rather than to use the logarithmic mean temperature difference between the fluid in the coil and the air-cooling coil, designers of air-cooled heat transfer surfaces have found it more convenient to express coil performance as a function of the initial temperature difference (ITD) between the fluid entering the coil and the air entering the coil (ambient air). Ignoring the subcooling effect, the iTD is identical to GTTD expressed in Equation [1] for direct systems and is equal to GTTD plus the cooling range for indirect systems.

Cheshire and Daltry (19) have developed an expression for the frontal area of the cooling coils which utilizes the initial air temperature difference between the fluid in the coil and the ambient air, and also takes into account the variations in height and depth of coolers, number of water passes, air flow, and water flow.

$$A_f = \frac{Q}{\Delta t}\left(\frac{1}{k_1 V_a^{\rho}} + \frac{pH}{k_2 n V_w} + \frac{1}{k_3 n U_c}\right) \qquad [7]$$

where: A_f = frontal area of cooling coils, $ft.^2$

Q = heat to be dissipated, Btu/hr.

Δt = maximum temperature difference between the water and the air, °F (which is the same as ITD)

V_a = air velocity at cooler, ft./sec.

ρ = air density, $lbs./ft.^3$

p = number of water passes

H = height of cooler, ft.

n = number of tube rows

V_w = water velocity in cooler, ft./sec.

U_c = over-all crossflow heat transfer coefficient, Btu/(hr. °F $ft.^2$)

k_1, k_2, k_3 = constants

Although calculations of heat transfer surfaces usually develop a logarithmic function, the linear function as derived by Cheshire and Daltry may well be accurate within the design limits of the dry-type cooling towers.

Dry cooling tower heat balance (20). In the system heat balance for a dry tower of the indirect type with a steam-electric generating plant, the heat transferred to the circulating water, the heat rejected to the air, and the heat rejected by the coil are all equal. The following are the basic formulas for these heat quantities:

Heat rejected to air:

$$GHR_a = W_a C_a (T_{a2} - T_{a1}) \qquad [8]$$

Heat rejected to circulating water:

$$GHR_w = W_w C_w (T_{w1} - T_{w2}) \qquad [9]$$

Heat rejected by finned heat exchanger:

$$GHR_c = U\ A\ LMTD\ F_g \qquad [10]$$

where: C_a = specific heat of air at constant pressure, Btu/(lb. °F)

C_w = specific heat of water, Btu/(lb. °F)

GHR_a = gross heat rejected to air, Btu/hr.

GHR_c = gross heat rejected by coil, Btu/hr.

GHR_w = gross heat rejected to circulating water, Btu/hr.

T_{a1} = temperature of air entering coil, °F

T_{a2} = temperature of air leaving coil, °F

T_{w1} = temperature of water entering coil, °F

T_{w2} = temperature of water leaving coil, °F

W_a = weight of air, lbs./hr.

W_w = weight of circulating water, lbs./hr.

F_g = crossflow correction factor, a function of air and water temperatures and pass arrangements, typically varies from 0.9 to 1.0 for large heat exchangers (21)

For a thermal-electric system in balance, it is obvious that:

$$GHR_a = GHR_w = GHR_c \qquad [11]$$

Solving simultaneous equations yields the following expression for gross heat rejected at the balance point, GHR_b:

$$GHR_b = \frac{ITD\,(e^z - 1)}{\frac{e^z}{W_a C_a} - \frac{1}{W_w C_w}} \qquad [12]$$

where: $$z = F_g\, U\, A \left(\frac{1}{W_a C_a} - \frac{1}{W_w C_w} \right) \qquad [13]$$

ITD = the initial temperature difference between the water entering the coil and the ambient air entering the coil, °F

Note the definition of ITD as used by Cates is different from that shown in Figure 10, which shows ITD as the initial temperature difference between the saturated temperature corresponding to the turbine back pressure, rather than the difference between the circulating water entering the coil and the air surrounding the coil. Smith and Larinoff (13) have used the definition of ITD for coil performance as the difference between temperature of saturated steam at turbine back pressure and ambient air, and this definition is generally used in European practice.

The numerical difference between the two methods of defining ITD is the subcooling of the circulating water below the saturated temperature of the exhaust steam at the turbine and can amount to approximately 3°F in practice. Either method of handling ITD is satisfactory as long as the subcooling effect is taken into account. The method used by Cates permits direct use of ITD without correction for subcooling effect on coil performance, and perhaps is the most logical method when viewed from a coil performance standpoint, whereas the other method would seem to be the better selection when considering over-all system performance because it relates tower performance directly to turbine back pressure. The foregoing example of different use of terms points up the need for standards of terms and definitions for the dry-type cooling tower industry; undoubtedly, such standardization will be forthcoming if dry-type cooling towers come into general use.

Cates expresses U as:

$$U = \frac{1}{\frac{1}{h'o} + \frac{A_r}{h_i} + \cdots\cdots + \frac{A_r}{hw}} \qquad [14]$$

where: $h'o$ = a collection of all conductances other than the inside coefficient, based on actual test data, Btu/(hr. ft.^{2} °F)

A_r = ratio of area of fin surface to inside area of tube

hw = tube wall conductance, Btu/(hr. ft.^{2} °F)

The equation for inside film heat transfer coefficient for water at ordinary temperatures is (18):

$$h_i = \frac{150(1 + .011\ t)\ V^{0.8}}{d^{0.2}} \qquad [15]$$

where: V = water velocity, ft./sec.

d = inside diameter of tube, inches

t = temperature of water, °F

Equation [15] is a simplification of the more general equation for liquids in fully developed turbulent flow given by McAdams (22) as:

$$h_c = .023 \left(\frac{Gd}{\mu_f}\right)^{.8} \left(\frac{\mu_f C_p}{K_f}\right)^{.33} \frac{K_f}{d} \qquad [16]$$

where: d = inside diameter, ft.

G = mass velocity, lbs./(hr. ft.^{2})

μ_f = absolute viscosity, lbs./(hr. ft.)

K_f = thermal conductivity of fluid, Btu/(hr. ft.^{2} °F/ft.)

C_p = specific heat at constant pressure, Btu/(lb. °F)

The inside tube transfer rate for isothermal condensing fluids such as steam is still controversial. The Heat Transfer Research Institute of Alhambra, California is embarked on an extensive experimental and correlation program for predicting isothermal and nonisothermal condensing rates. The methods now in common usage for isothermal condensation are the method of Dukler (23), the method of Kirkbride (24) — which are modified Nusselt correlations — and the method of Akers (25) which takes into account the shear effect on the condensate caused by the velocity of the uncondensed vapor (12).

Effectiveness—N_{tu} Approach

The technique of arriving at an optimum heat exchanger design is a complex one due to the mathematics involved. Even more significant, however, are the many qualitatave judgements that must be introduced into the analysis. The variables previously described are complex functions that do not lend themselves to ease of evaluation. Consequently, except for simple configurations, model tests generally are used to establish their effect in a given cooling element.

Kays and London (17) describe another approach to heat exchanger design in terms that allow a better visualization of the interaction of various major parameters on the efficiency of a heat exchanger. They describe exchangers in terms of "effectiveness" and "number of heat transfer units".

The "effectiveness" term defines the heat transfer performance. This term compares the actual heat transfer rate to the maximum possible heat transfer rate and is a measure of the heat transfer effectiveness of the cooling element.

$$E = \frac{W_w C_{pw} (T_{w_{in}} - T_{w_{out}})}{W_a C_{pa} (T_{w_{in}} - T_{a_{in}})} \qquad [17]$$

where: E = exchanger heat transfer effectiveness, nondimensional

W_w = water flow rate, lbs./hr.

W_a = air flow rate, lbs./hr.

C_{pw} = specific heat of water, lbs./hr.

C_{pa} = specific heat of air, Btu/(lb. °F)

T_w = temperature of water, °F

T_a = temperature of air, °F

The number of heat transfer units, N_{tu}, is a nondimensional expression of the heat transfer size of the exchanger.

$$N_{tu} = \frac{A\,U}{W_a\,C_{pa}} \qquad [18]$$

When the N_{tu} is small, the exchanger effectiveness is also small. It is apparent from an examination of Equation [18] that the cost of attaining a large N_{tu} and, consequently, a high degree of effectiveness is tied closely to the capital investment required to provide a large heat transfer area or an improvement in the conductance, U.

The relationship between E and N_{tu} for a crossflow cooling situation with the air assumed to be mixed is shown in Figure 11, taken from (17). The effectiveness of a heat transfer element increases sharply with a greater number of heat transfer units until the curve levels out and becomes almost asymtotic. Considerable expense is required to obtain the last 20 to 30 percent of effectiveness. Therefore, the optimum cost-effective cooling element may not be the most efficient one.

Theory of Thermodynamic Cycles

The Carnot cycle. The understanding of the basic Carnot cycle is useful in studying the improvement of the Rankine cycle, which is described in subsequent paragraphs. The Carnot cycle is shown in Figure 12, plotted on a temperature-entropy diagram.

The simplest statement of the second law of thermodynamics is that heat will not flow of its own accord from a cold body to a hot body. The second law may be rephrased to state that not all of a given quantity of heat can be converted into useful work.

The Carnot cycle, comprising two constant-entropy processes and two constant-pressure processes, all of which are reversible, is the most efficient power plant cycle conceivable. Temperature 2 - 3 is the maximum temperature available to the cycle and temperature 1 - 4 is the lowest temperature available.

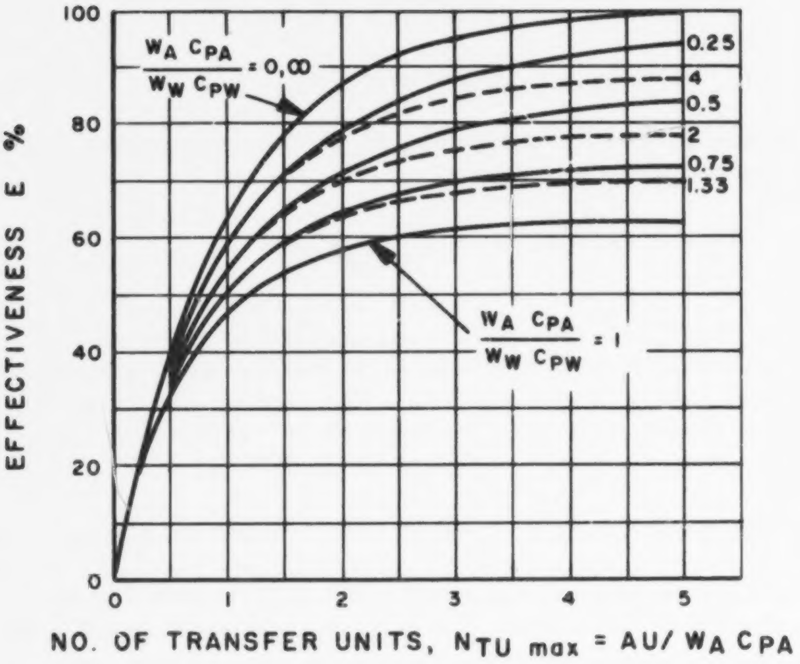


FIGURE 11 — HEAT TRANSFER EFFECTIVENESS AS A FUNCTION OF NUMBER OF TRANSFER UNITS (N_{TU}) CROSSFLOW EXCHANGER WITH AIR MIXED(17)

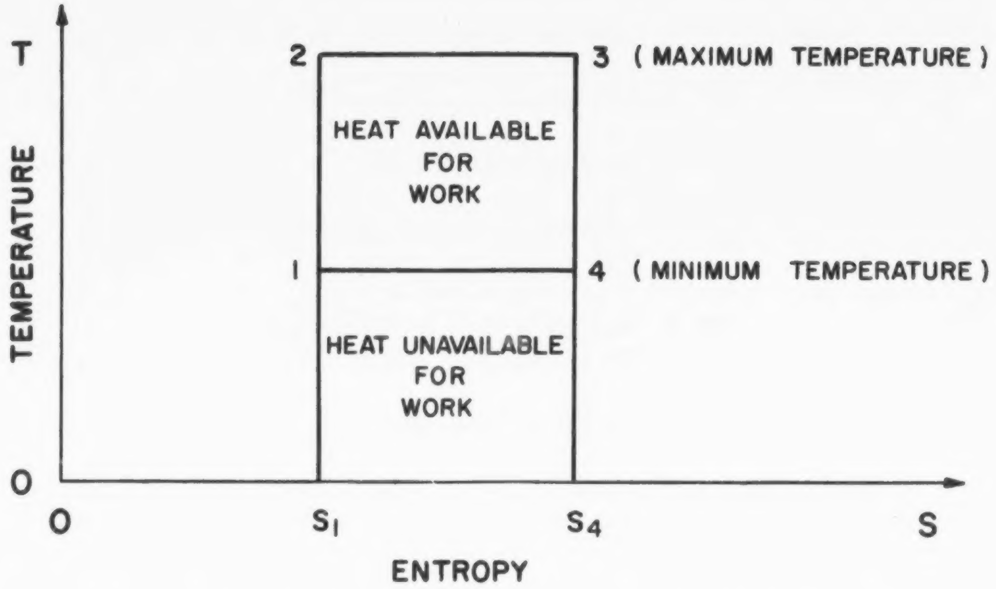

T
TEMPERATURE
2
3 (MAXIMUM TEMPERATURE)
HEAT AVAILABLE FOR WORK
1
4 (MINIMUM TEMPERATURE)
HEAT UNAVAILABLE FOR WORK
0
0
S_1
S_4
S
ENTROPY

FIGURE 12 — CARNOT CYCLE PLOTTED ON TEMPERATURE— ENTROPY DIAGRAM

In power plant practice, temperature 1 - 4 is the temperature of the circulating water, or, in the case of an air-condensing system, the ambient air temperature.

The thermal efficiency of the Carnot cycle is expressed as follows:

$$\text{Carnot cycle efficiency} = \frac{\text{heat available for work}}{\text{total heat supplied}} =$$

$$\frac{(T_2 - T_1)\,(S_4 - S_1)}{(T_2 - T_0)\,(S_4 - S_1)} = \frac{T_2 - T_1}{T_2} \qquad [19]$$

The Carnot cycle represents the highest possible efficiency of a cycle. Although such efficiency is unobtainable on a practicable basis, the cycle provides a basis for measuring the efficiencies of power systems.

The Rankine cycle. The general energy equation expresses the first law of thermodynamics as it applies to steady-flow processes, such as apply to the steam power plant cycle, as (26):

$$PE_1 + KE_1 + WH_1 + Q = PE_2 + KE_2 + WH_2 + Wk \qquad [20]$$

where: PE = potential energy, ft-lb. or Btu

KE = kinetic energy, ft-lb. or Btu

H = total enthalpy, Btu/lb.

Q = heat transferred to or from the system, Btu

Wk = work done on or by the system, Btu

W = weight of fluid, lbs.

The heat balance for the power plant assumes the cycle to be a closed system. Changes in potential energy are not significant and by treating the power plant components as integral units, changes in kinetic energy do not have to be considered. The general energy equation for a steam power plant, then, is written as:

$$Wk = W(H_1 - H_2) + Q \qquad [21]$$

where: H_1 = enthalpy of entering steam or water, Btu/lb.

H_2 = enthalpy of leaving steam or water after expansion, Btu/lb.

Q = heat added to the system between conditions 1 and 2, Btu/hr.

Wk = work done on or by the system between conditions 1 and 2, Btu/hr.

W = flow of steam or water, lbs./hr.

Figure 13 shows the diagram of an elementary steam plant cycle known as the Rankine cycle.

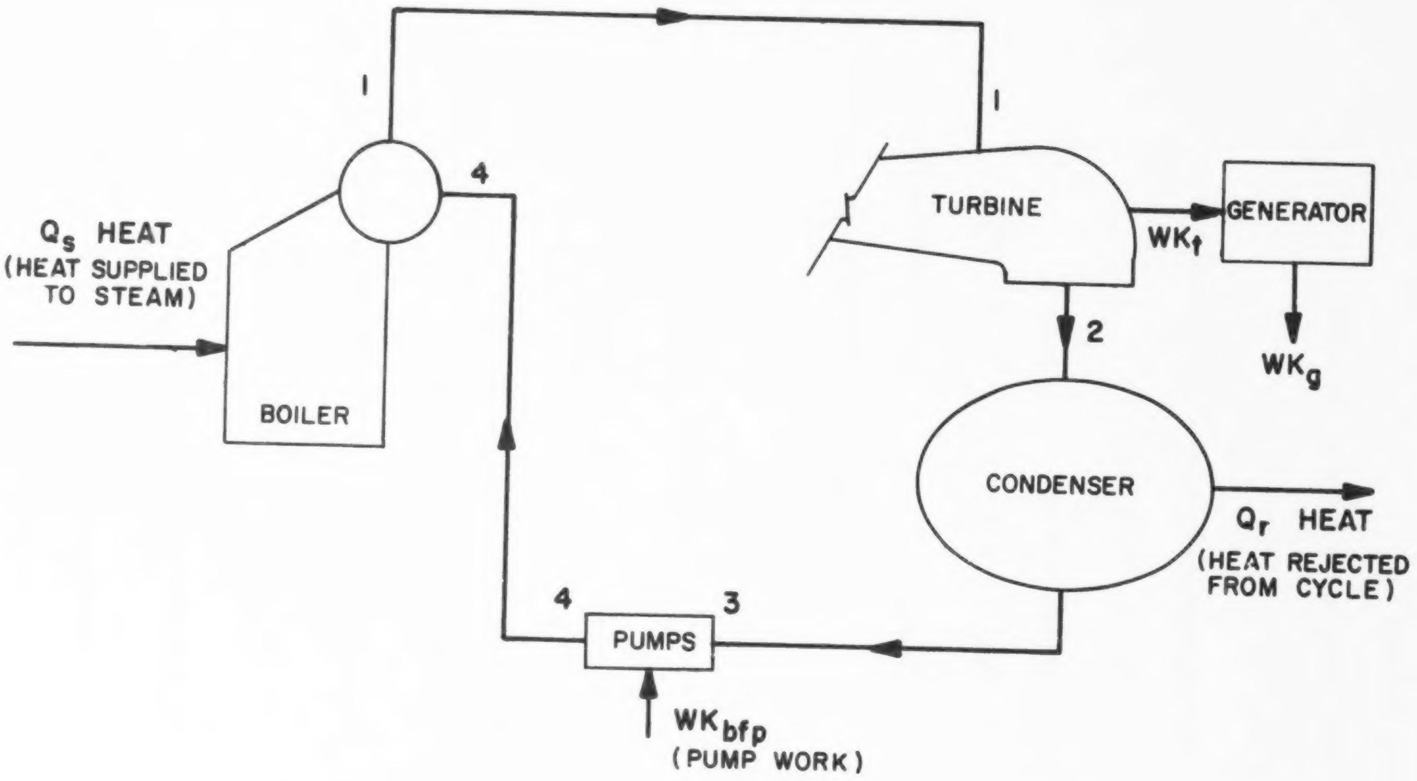


FIGURE 13 — DIAGRAM OF RANKINE CYCLE

Referring to Figure 13,

a. Heat added to the cycle by the boiler:

$$Q_s = W(H_1 - H_4) \tag{22}$$

b. Heat rejected from the cycle by the condenser:

$$Q_r = W(H_2 - H_3) \tag{23}$$

c. Work done by turbine:

$$Wk_t = W(H_1 - H_2) = W\eta_t(H_1 - H_{2'}) \tag{24}$$

where: η_t = over-all turbine efficiency

$H_1 - H_{2'}$ = total isentropic available energy, Btu/lb.

d. Generator output:

$$Wk_g = W\eta_g(H_1 - H_2) = \eta_g Wk_t \tag{25}$$

where: η_g = over-all generator efficiency

e. Boiler feed pump work:

$$Wk_{bfp} = W(H_4 - H_3) \tag{26}$$

f. Thermal efficiency based on gross generator output:

$$\text{Gross thermal efficiency} = \frac{Wk_g}{Q_s} \tag{27}$$

g. Thermal efficiency based on net generator output:

$$\text{Net thermal efficiency} = \frac{Wk_g - Wk_{bfp}}{Q_s} \tag{28}$$

h. Gross turbine cycle heat rate:

$$\text{Gross heat rate} = \frac{3{,}413}{\text{gross thermal efficiency}} =$$

$$\frac{3{,}413\ Q_s}{Wk_g} = \frac{Q_s}{KW_g} \qquad [29]$$

i. Net turbine cycle heat rate:

$$\text{Net heat rate} = \frac{3{,}413}{\text{net thermal efficiency}} =$$

$$\frac{3{,}413\ Q_s}{Wk_g - Wk_{bfp}} = \frac{Q_s}{KW_g - KW_{bfp}} \qquad [30]$$

j. Plant net heat rate, taking into account boiler efficiency and auxiliary power requirements:

$$\text{Net plant heat rate} = \frac{Q_s}{\eta_b (KW_g - KW_{bfp} - KW_a)} \qquad [31]$$

where: η_b = boiler efficiency

KW_g = generator output at generator terminals, kw

KW_{bfp} = boiler feed pump power, kw

KW_a = auxiliary power (excluding boiler feed pump), kw

Wk_g = generator work, Btu/hr.

Wk_{bfp} = boiler feed pump work, Btu/hr.

In modern power plant design, the regenerative reheat cycle is used rather than the Rankine cycle. The modern regenerative reheat cycle, however, is but an improvement of the basic Rankine cycle and, therefore, an understanding of the simple Rankine cycle is essential.

Improvements to the Rankine cycle. The basic efficiency of the Rankine cycle can be improved by increasing the temperature of the steam from the boiler by superheating, by reheating the steam to its maximum temperature after it has performed a certain amount of work in the turbine, and by means of regenerative feedwater heating.

The regenerative reheat cycle is used with all large, modern steam-generating plants. Figure 14 shows the basic diagram of a regenerative reheat turbine cycle.

The equation for the heat rate of the regenerative reheat turbine cycle, more commonly referred to as the reheat turbine cycle, is:

$$\text{Gross heat rate} = \frac{W_t (H_t - h_{fw}) + W_{rhtr} (H_{hr} - H_{cr})}{\text{generator output}} \qquad [32]$$

where: W_t = throttle flow, lbs./hr.

W_{rhtr} = reheater flow, lbs./hr.

H_t = throttle enthalpy, Btu/lb.

h_{fw} = final feedwater enthalpy, Btu/lb.

H_{hr} = enthalpy leaving reheater, Btu/lb.

H_{cr} = enthalpy entering reheater, Btu/lb.

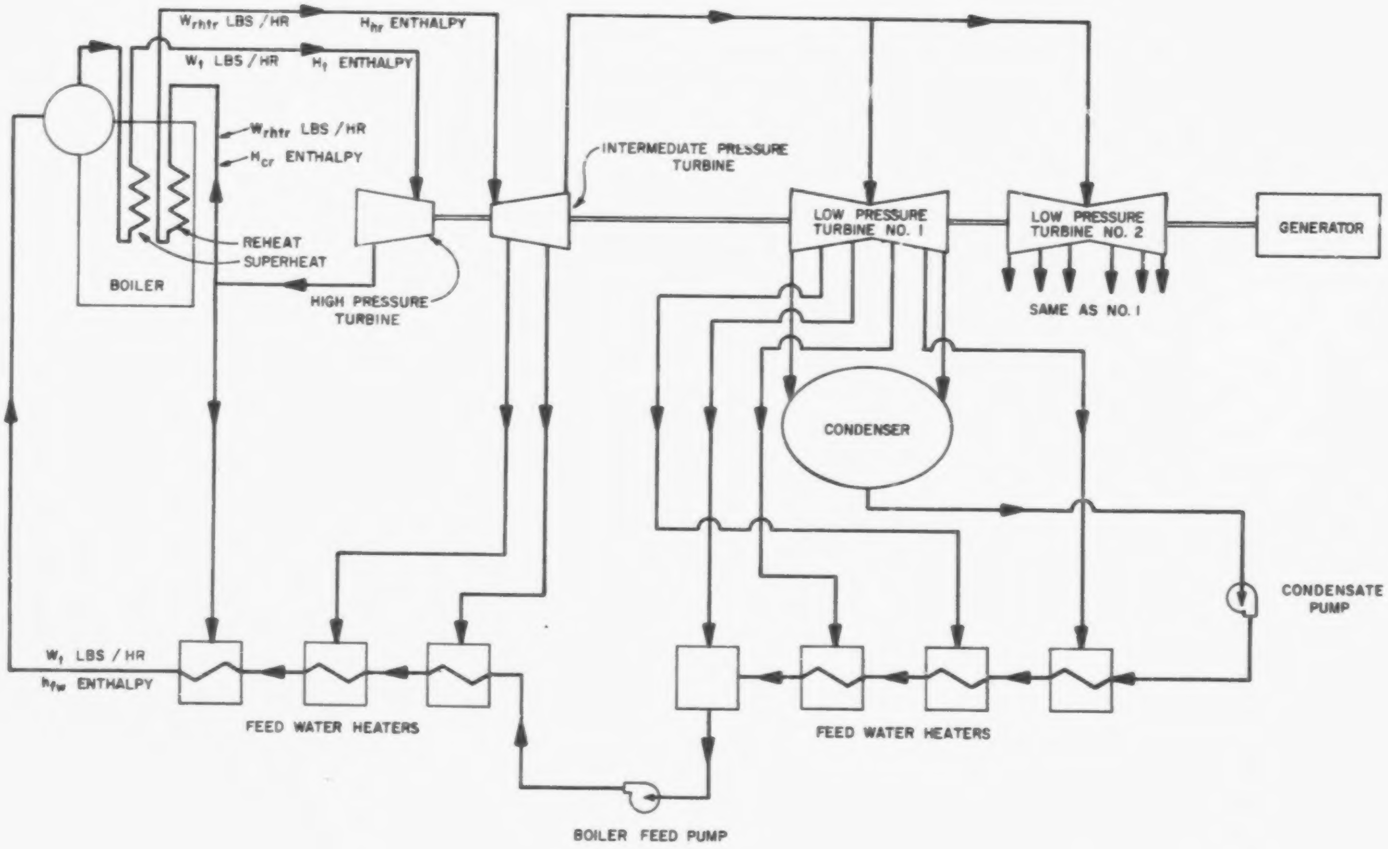


FIGURE 14 — TYPICAL FLOW DIAGRAM FOR REGENERATIVE REHEAT CYCLE

SECTION III

PERFORMANCE

Performance of Dry-Type Cooling Towers

The concept of initial temperature difference (ITD), discussed in Section II under "Theory of Heat Transfer from Air-Cooled Coils" and as illustrated in Figure 10, is essential in understanding the performance of a dry-type cooling system under varying ambient air temperatures and turbine loads. In this report, the ITD is considered to be the difference between the saturated steam temperature corresponding to the turbine back pressure at the exhaust flange and the ambient air temperature, since this method directly indicates the effect of variations in ambient air temperature upon turbine performance. In adopting this definition of ITD, there must be compensation for condensate subcooling in the indirect-type cooling system and for steam-pressure loss in the exhaust steam trunk for the direct-type condensing system when considering cooling coil performance. Depending upon the design and performance of the system, there may be a difference of from one-half degree to several degrees between the ITD as defined above and the ITD defined as the difference between the temperature of either warm circulating water entering the coils in the indirect system or the temperature of condensing steam in the direct system and the ambient air temperature.

As discussed in the section on theory, the actual heat transfer coefficient, U, for a cooling element in a dry-type tower is designed on a trial basis with a testing program to establish or verify the design factors. From data obtained on existing and proposed natural-draft installations, the U factor varied from 184 to 238 Btu/(hr. $^{\circ}$F ft.^{2}) per row and averaged 202 Btu/(hr. $^{\circ}$F ft.^{2}) per row on a frontal-area basis. The value would be less if calculated on a total cooling surface area. No indirect, mechanical-draft installations of any size for generating units have been constructed, so their U factor must be estimated on a theoretical basis. It would appear that the pressure drop across the cooling element in a mechanical-draft tower would be of less concern than in a natural-draft tower where a slight increase in the pressure drop would increase the tower height by many feet with a consequent increase in construction cost. In a mechanical-draft tower, an increase in pressure drop would be offset by increasing fan horsepower. Therefore, it is likely that the cooling elements could be closer together in a mechanical-draft tower and the U factor on a frontal-area basis would be higher than for a natural-draft tower—say in the range of 300 to 350 Btu/(hr. $^{\circ}$F ft.^{2}) per row.

An analysis of the heat transfer effectiveness "E" versus heat transfer size "N_{tu}" indicates that existing units are designed for far less than the ideal maximum heat transfer effectiveness due to cost considerations in developing a cooling system for the lowest expenditure of capital and operating expense.

The heat rejection performance of the tower and the thermodynamic performance of the turbine are the two most significant factors in the operation of a dry-type condensing system. The complex relationships which exist between the tower and the turbine must be determined in order to predict the performance of a combination of turbine-generator and dry-type cooling tower.

Since the performance of a dry-type cooling tower system and the turbine which it serves are so closely related, the complete condensing system (cooling coils, method of moving air, pumps, piping, condenser) and turbine can best be considered as one integral unit in studies of economic comparisons of various systems of a dry tower for a specific turbine.

The performance data and prices for dry-type coils used in the economic evaluation of this report were furnished by the Hudson Products Corporation for mechanical-draft towers and by Dr. Heller for natural-draft towers, and represent heat exchangers actually being offered to the utility industry by established manufacturers.

Certain of the heat exchange data supplied by the manufacturers who cooperated in this research are of a proprietary nature. This proprietary information was incorporated into our analyses and is reflected in the results shown herein; however, these data are not included in our report in their original form, but may be available by direct inquiry to the original sources.

Natural-draft towers. During conferences with Dr. Heller in Budapest, basic information was obtained regarding the performance of Heller-type towers which shows the relationship between heat transfer, flow of air and water, natural-draft tower height, and other factors. This information is shown on the following generalized curves.

Figure 15 shows the relative coil performance in heat rejection along with the relative water- and air-pressure losses through the heat exchanger coils. The range of values reflected by the curves of Figure 15 are as follows:

Air flow – 400,000 to 1,000,000 pounds per hour per column

Air-pressure drop through coils – 0 to 0.4 inches head loss (water gauge)

Water flow – 130,000 to 260,000 pounds per hour per column

Water head loss through coils – 0 to 50 feet pressure head

Heat transfer per 20-meter heat exchanger column – 0 to 100,000 Btu per hour per °F.

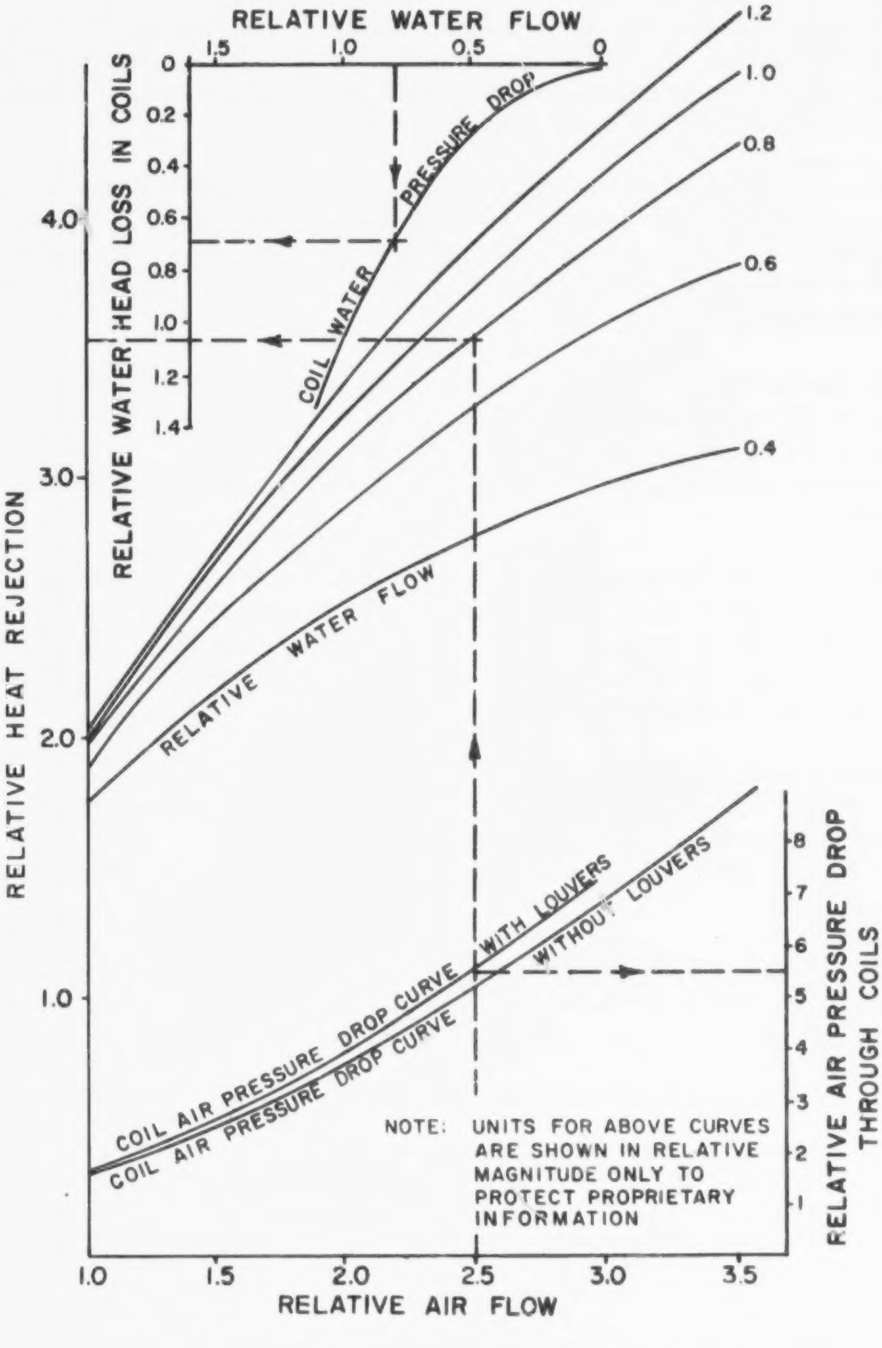


FIGURE 15 — COIL PERFORMANCE VERSUS AIR AND WATER FLOW

Figure 16 illustrates the relative coil performance as related to the relative height of the natural-draft tower, water flow and ITD. The range of values reflected by the curves of Figure 16 are as follows:

Air flow – 400,000 to 1,000,000 pounds per hour per column

Water flow – 90,000 to 300,000 pounds per hour per column

Tower height – 0 to 400 feet

Heat transfer per 15-meter heat exchanger column – 0 to 90,000 Btu per hour per °F.

These curves are based on a combination of basic theory and proprietary data developed in Hungary, so they were analyzed to determine their applicability to conditions in the United States.

When air inside a natural-draft cooling tower is heated by the coils, a draft is created, causing an upward flow of air. At some flow of air, tower conditions reach an equilibrium where the draft created by the heated air matches draft losses caused by the flow of air. The equation for theoretical stack effect of heated air, as taken from (27), is:

$$D = .256\, h\, p' \left(\frac{1}{T_a} - \frac{1}{T_g} \right) \qquad [33]$$

where: D = draft, inches H_2O

h = effective stack height, ft.

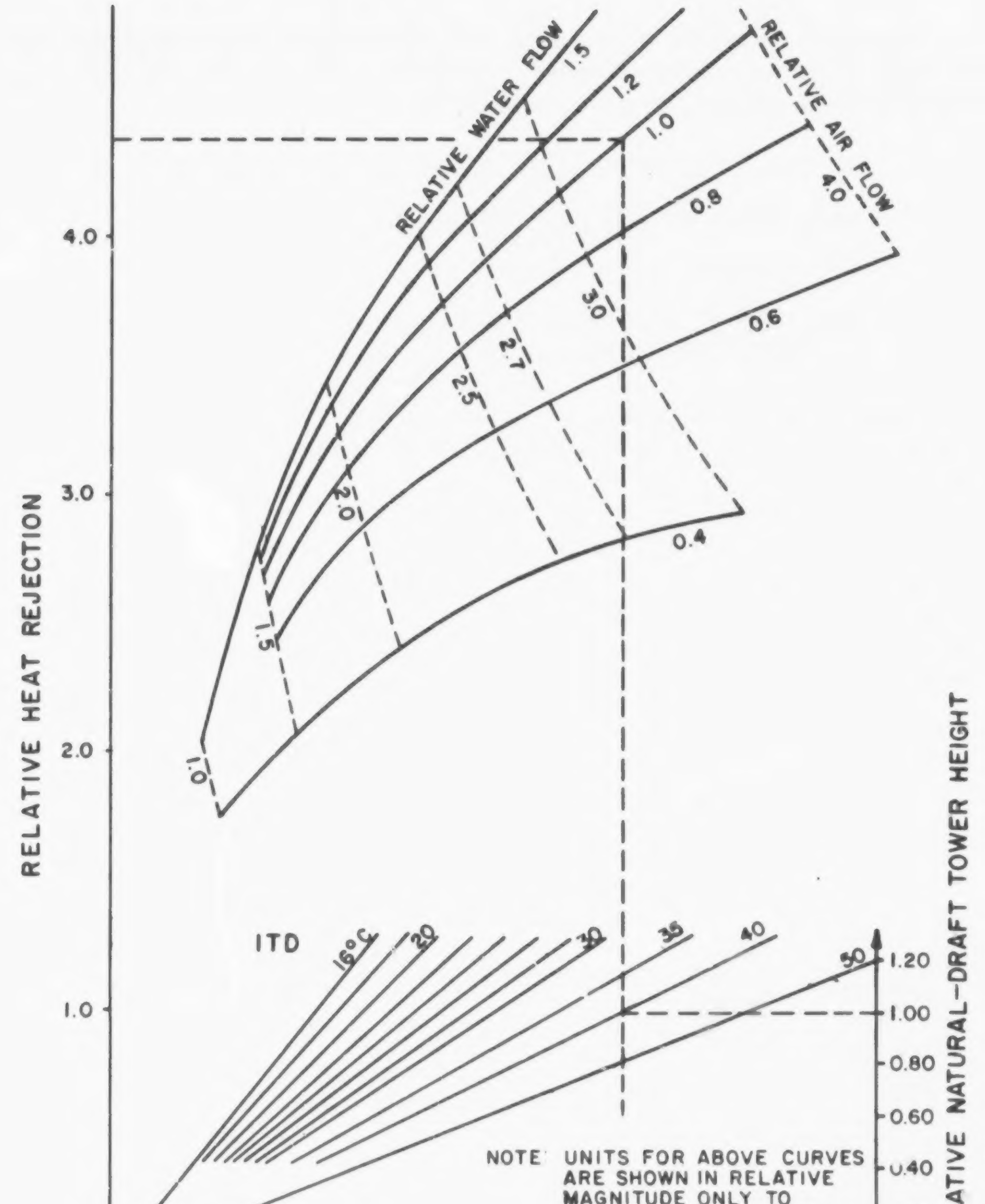
p' = atmospheric pressure, inches Hg

T_a = ambient air temperature, $^{\circ}R$

T_g = inside tower air temperature, $^{\circ}R$

The effective tower height must be corrected for a portion of the coil height, elevation of the tower, ambient air temperature, and variation in atmospheric pressure due to tower height.

Draft losses may be grouped into three categories: 1) draft loss across the coil, 2) exit loss (27), and 3) draft losses within the tower. The heat transfer coil used is a key factor in tower design. The heat transfer rate, water and air flow rates, and water and air pressure drops through and across the coil are interrelated. For purposes of this study, information on coils developed by Dr. Heller, as shown on Figure 15, was used.



NOTE: UNITS FOR ABOVE CURVES ARE SHOWN IN RELATIVE MAGNITUDE ONLY TO PROTECT PROPRIETARY INFORMATION

FIGURE 16 — COIL PERFORMANCE VERSUS WATER FLOW, TOWER HEIGHT AND INITIAL TEMPERATURE DIFFERENCE (ITD)

The exit loss is a function of the exit velocity of the heated air as it leaves the tower. The exit velocity is determined by the volume of discharged air and the upper tower diameter. There are several alternatives for estimating exit loss. In this study, the exit velocity was considered a function of the air flow per cooling element. Empirical data supplied by Dr. Heller was used.

The draft losses within the tower consist of losses due to frictional resistance and changes in tower cross section. These losses are small in comparison with the coil and exit losses.

Mechanical-draft towers. The sizing of mechanical draft, either forced or induced, is much simpler than for natural draft. Since the required air flow is assured by the fans, the principal concern is the characteristics of the cooling coils and the pressure loss across them.

For purposes of this study, mechanical-draft data supplied by Hudson Products Corporation was utilized. Some of the information received was proprietary in nature and, consequently, only a portion of it is illustrated. Figure 17 shows the effect of cooling tower size upon turbine back pressure for a range of ambient air temperatures. The number of cooling units ranges from approximately 15 to 45 for a cooling range of 45°F to 15°F for an 800-mw fossil-fueled unit. From these curves, it is readily recognized that the use of 15 cooling units would result in extremely high turbine back pressure in a location subject to high ambient air temperatures. On the other hand, 45 cooling units would provide extremely low back pressures but at a considerably higher cooling tower investment.

Figure 18 illustrates the power requirements necessary to drive the fans and pumps over a range of ITD's from 30°F to 80°F.

Variations in altitude will not affect tower configuration, but fans and drive mechanisms must be varied in size to move the appropriate mass of air.

Tower performance for varying load and ambient air temperatures. The optimum size and cost of natural- and mechanical-draft towers was established with the analyses described above. For the economic optimization of towers at a specific site, it was necessary to consider their operation at part loads and with the full range of temperatures that they would experience. Therefore, it was necessary to develop a means of evaluating their performance under these conditions.

From an analysis of performance data of natural-draft, dry-type cooling towers that have been designed in Europe, the relationship between ITD and heat rejection for typical natural-draft towers can be determined.

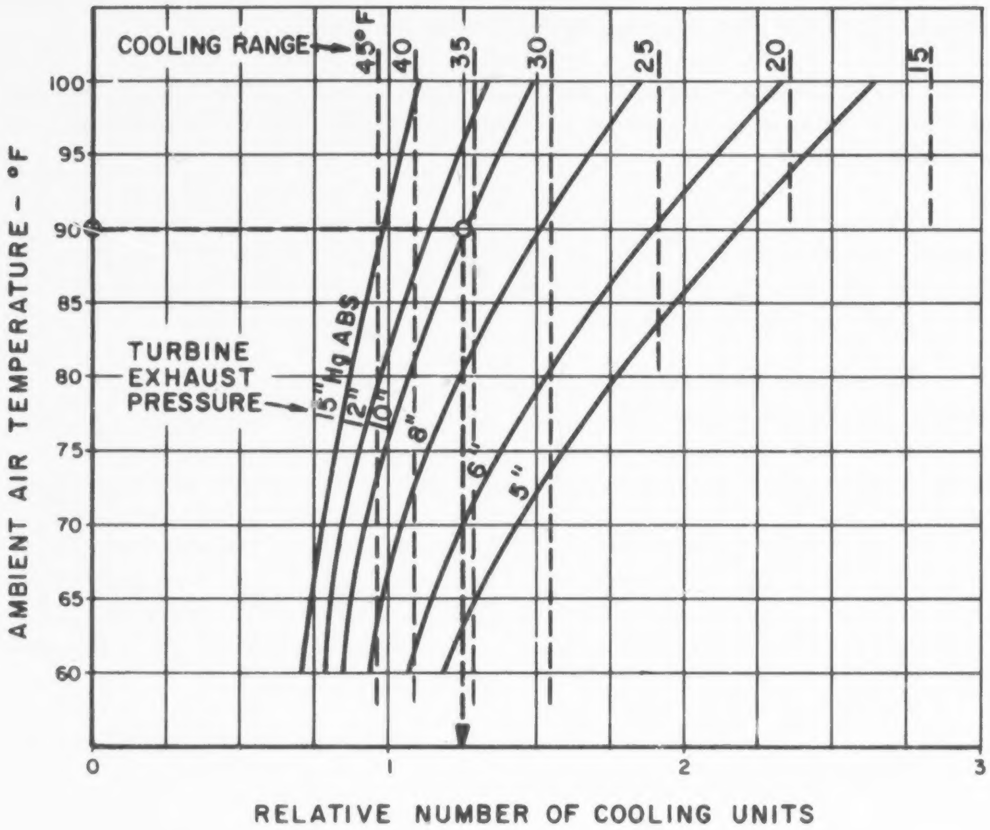


FIGURE 17—COOLING UNITS REQUIRED FOR MECHANICAL-DRAFT, DRY-TYPE COOLING SYSTEM

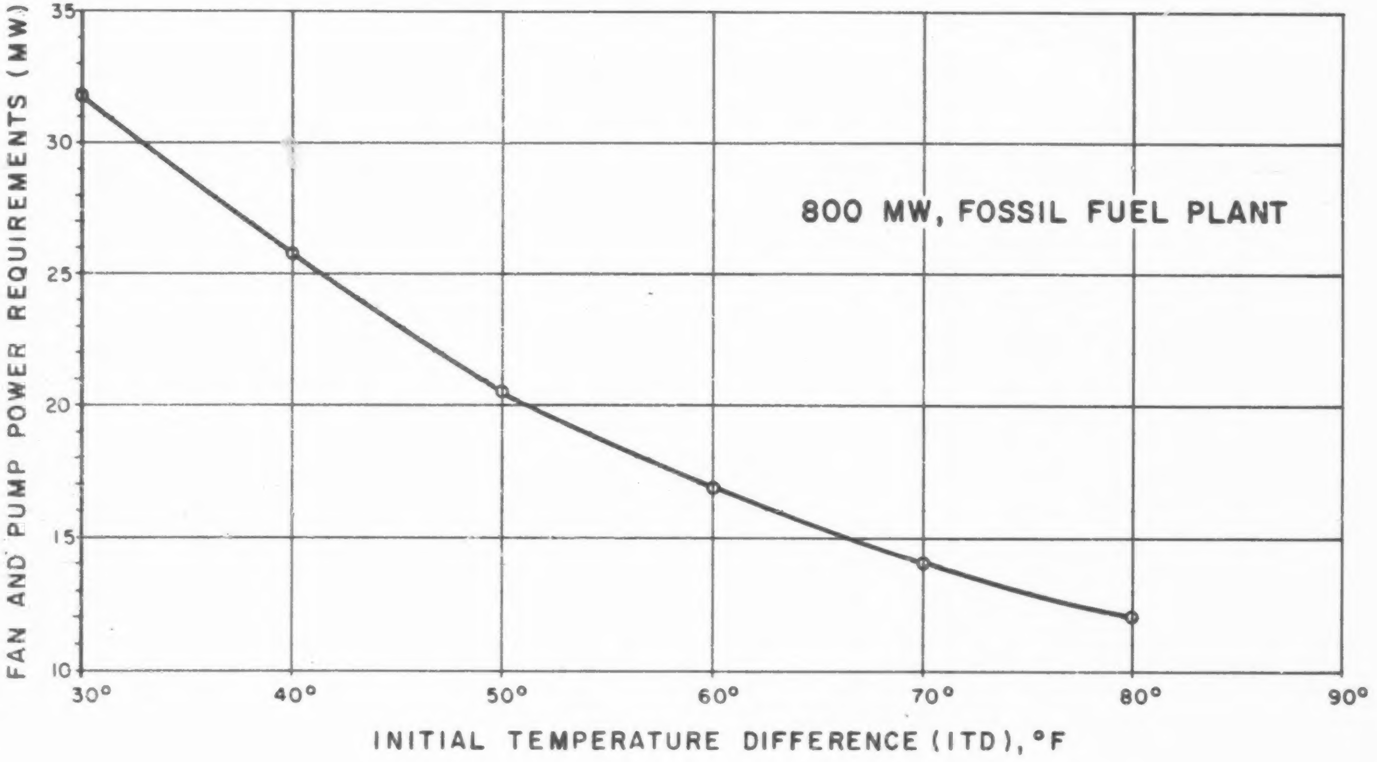


FIGURE 18—CURVE OF FULL LOAD AUXILIARY POWER REQUIREMENTS VERSUS ITD—MECHANICAL-DRAFT, DRY-TYPE COOLING SYSTEM

The performance of natural-draft towers was found to be reasonably expressed by the equation:

$$\text{ITD} = A\,Q^{b} \qquad [34]$$

where: ITD = initial temperature difference, °F

Q = heat rejection, 10^6 Btu/hr.

A = tower constant

b = constant, depending on natural- or mechanical-draft towers

The performance of four existing or designed natural-draft towers was analyzed by computer with the results shown below.

For the Ibbenbüren and Rugeley plants, which have been built and are in operation:

Ibbenbüren (150 mw)	ITD	$= 0.501\,Q^{.717}$
Rugeley (120 mw)	ITD	$= 0.247\,Q^{.793}$

From information received from M.A.N. (Maschinenfabrik Augsburg-Nürnberg) for a 200-mw, indirect, dry-type cooling system, designed but not built, using two different types of cooling coils:

"A" coils	ITD	$= 0.410\,Q^{.762}$
"B" coils	ITD	$= 0.544\,Q^{.730}$

The four exponents of the equations as found above are similar, ranging from 0.72 to 0.79 with an average of 0.75.

Figure 19 shows the plot of the equation ITD = A Q^b for the four natural-draft towers analyzed. Note that the approximate ITD for the Rugeley Station is 35°F; for Ibbenbüren, 50.5°F; and for the 200-mw station with the "A" coil and "B" coil, 80°F.

The performance curves published by Smith and Larinoff (13) indicate that the relationship between ITD and heat rejection for a mechanical-draft dry tower is approximately linear. The operating curves of the Neil Simpson plant at Wyodak, Wyoming, furnished by GEA, Gesellschaft für Luftkondensation, also indicate a near linear relationship for heat rejection versus ambient air temperature for a fixed

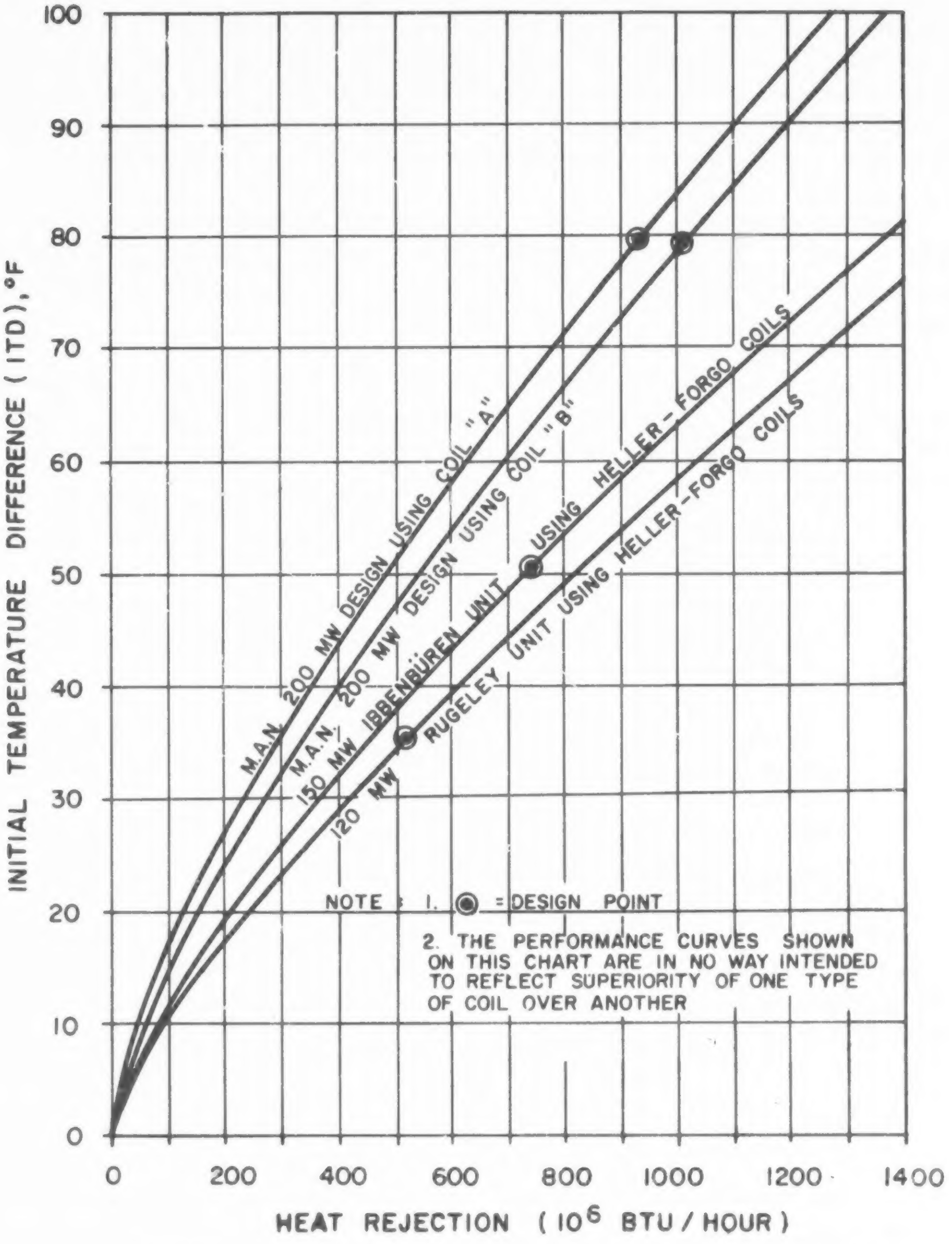


FIGURE 19— NATURAL-DRAFT, DRY-TYPE TOWER PERFORMANCE CAPABILITY WITH VARIATION OF INITIAL TEMPERATURE DIFFERENCE

turbine back pressure (saturated steam temperature), Figure 20. Since ambient air temperature, used as the abscissa in Figure 20, is equal to the saturated steam temperature minus ITD, the ambient air temperature varies inversely as the ITD for a fixed saturated steam temperature (turbine back pressure as used in the figure). Therefore, the slope of the curves on Figure 20 can also be considered to represent ITD versus heat rejection. Appendix A, in the description of the Volkswagen plant, describes how Figure 20 is used as a guide for control of the cooling system.

Based on the above information, the exponent "b" in equation [19] was established at 0.75 for natural-draft and 0.91 for mechanical-draft towers to determine part-load operation and to evaluate the effects of the variation in temperature during the year.

The above findings as to the 0.75 exponent for natural-draft towers corresponds with the statement in (28), the only publication found that had any reference to natural-draft, dry-type cooling tower performance. Some of the approximating rules it contains are:

> The ITD is a function of heat rejection to somewhat more than the 2/3 power.
>
> The air mass flow is a function of heat rejection to the 1/3 power.
>
> The rise in air temperature is a function of heat rejection to the 2/3 power.
>
> The air flow becomes less at partial heat rejection load because the chimney action of the tower is decreased.

The somewhat more than 2/3 exponential curve for ITD seems to be reasonably close to our 0.75. It is also stated in (28) that, with mechanical-draft towers, the ITD is approximately proportional to the heat rejection of the tower. This information is in accordance with the curves published in (13) and as shown in Figure 20.

The design point for the dry-type cooling tower system for the Rugeley Station is reported to be 1.3 inches Hg turbine back pressure at 52°F ambient air temperature. Since the saturated steam temperature corresponding to 1.3 inches Hg is 87°F, the ITD is 35°F (87°F −52°F). The design point could have been taken at 57°F ambient air temperature and 1.5 inches Hg turbine back pressure, since this condition also represents an ITD of 35°F.

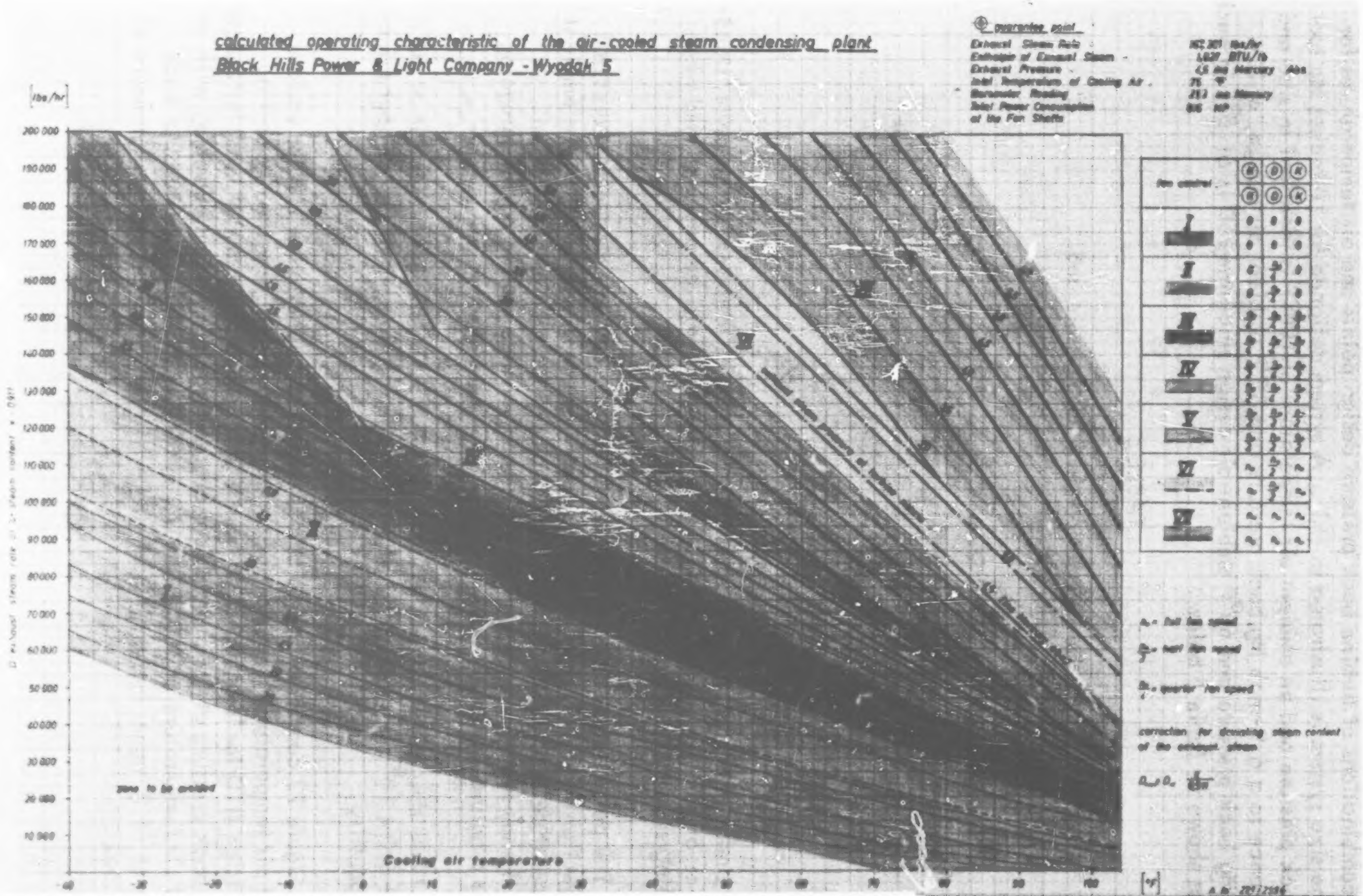


FIGURE 20—GRAPH OF CALCULATED OPERATING CHARACTERISTICS (PREDICTED PERFORMANCE) FOR DIRECT, AIR-COOLED CONDENSING SYSTEM, NEIL SIMPSON PLANT, WYODAK, WYOMING (FROM GEA)

Design ITD. The fact that a dry-type cooling tower system can have a number of combinations of turbine back pressure design points and air temperatures for the same size system is illustrated in (13). A system designed for 10 inches Hg turbine back pressure and an ambient air temperature of 100°F is identical in size and performance to a 6.9-inch Hg back pressure and 85°F ambient air design, or 3.6 inches Hg back pressure and 60°F ambient air design since the ITD is 61.5°F in each case, as shown in Table 5 below.

TABLE 5

Possible Variations in Back Pressure and
Ambient Air for a Given ITD

Turbine Back Pressure (inches Hg)	Saturated Steam Temperature (°F)	Air Temperature (°F)	Design ITD (°F)
10.0	161.5	100	61.5
6.9	146.5	85	61.5
3.6	121.5	60	61.5

Undoubtedly, in order to forestall confusion as to tower performance, standardization of back pressure and air temperature design will be accomplished when air cooling systems are more prevalent.

The performance of the dry tower for various ambient air temperatures and heat rejection loads can be illustrated by performance curves in which the ambient air temperature is plotted against turbine back pressure and heat rejection. Figure 21 shows typical curves of dry tower performance plotted on the foregoing basis for a natural-draft tower.

Note that this tower has a dual rating 4.0×10^9 Btu heat rejection at 50°F ITD and 6.0×10^9 Btu at 67.8°F, illustrating the relationship of ITD to heat rejection capability.

Figure 22 compares turbine back pressure as a function of ITD for various ambient air temperatures. This set of curves shows clearly how the dry cooling tower design (ITD for heat rejection from a turbine operating at full load) influences turbine back pressure at various ambient air temperatures. As an example, with an ambient air temperature of 100°F, an ITD of 80°F will result in a turbine back pressure of 15.3 inches Hg; an ITD of 40°F will result in a turbine back pressure of 5.89 inches Hg.

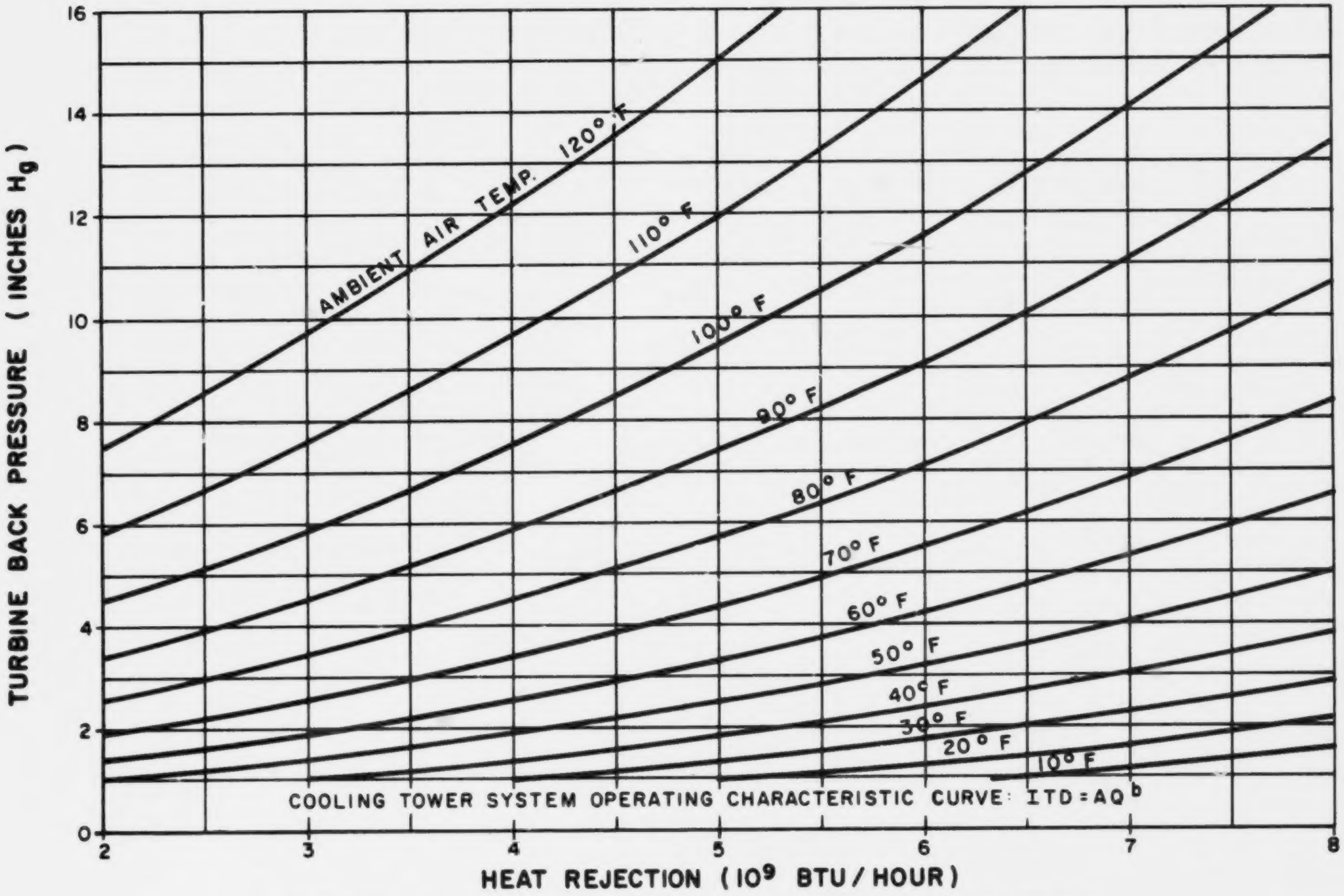


FIGURE 21—NATURAL-DRAFT, DRY-TYPE COOLING TOWER OPERATING CHARACTERISTICS

(ITD=50°F AT 4.0 x 10^9 BTU/HR & ITD=67.8°F AT 6.0 x 10^9 BTU/HR)

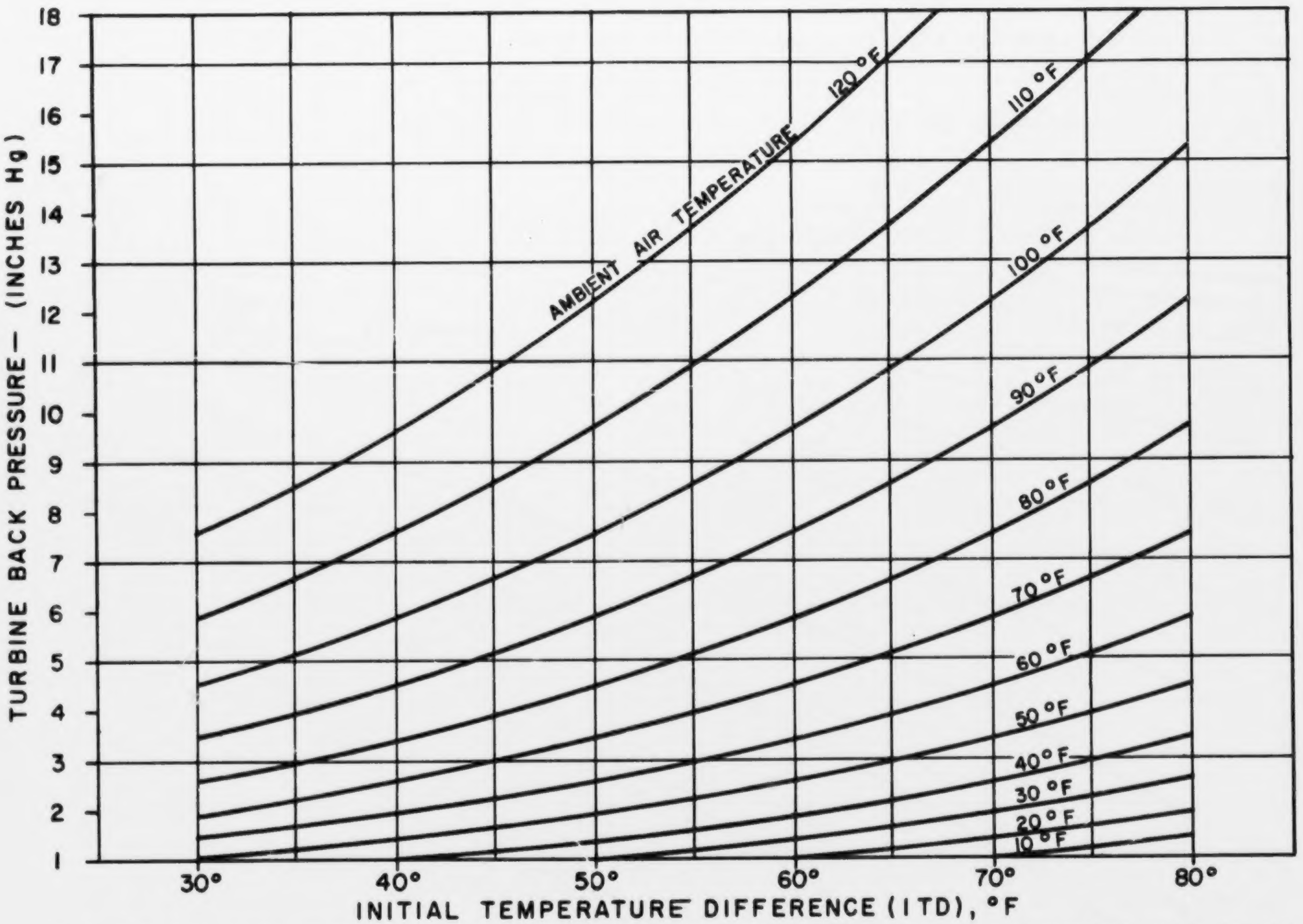


FIGURE 22—DRY-TYPE COOLING TOWER SYSTEM: TURBINE BACK PRESSURE VARIATION WITH INITIAL TEMPERATURE DIFFERENCE (ITD) FOR GIVEN AMBIENT AIR TEMPERATURES

Performance of Turbine Used With Dry-Type Cooling Towers

An explanation of the turbine expansion line and the available energy of the steam as it flows through the turbine is useful in understanding the effect of back-pressure variation on turbine efficiency. The Mollier Chart is a plot of steam properties made from steam tables where enthalpy is plotted against entropy, with other parameters such as temperature, degrees of superheat and percent moisture also shown.

Figure 23 shows how the expansion line for a reheat turbine would appear plotted on a Mollier Chart.

Turbine throttle steam at enthalpy H_t expands through the high-pressure turbine from point 1 to point 2, returns to the boiler at reheater pressure where the steam temperature is raised to the hot reheat temperature (generally the same as the initial throttle temperature), flows to the intermediate section of the turbine at enthalpy H_{hr} at point 3, and then expands through the intermediate- and low-pressure turbines to the exhaust pressure.

Figure 23 shows the steam expanding to two exhaust pressures, "A" and "B", which will serve to illustrate the change in efficiency and turbine capability with an increase in back pressure.

Under the first assumed operating condition, the steam expands to point "A", 2 inches Hg back pressure. In passing through the intermediate-pressure and low-pressure turbines, each pound of steam does work equivalent to $H_3 - H_5$ (with appropriate correction made for the exhaust losses of the exhaust steam which are unavailable as work). If the exhaust pressure is raised from 2 inches Hg to 5 inches Hg, less work is accomplished by each pound of steam by the amount $H_4 - H_5$ Btu per pound (also corrected for exhaust losses for 5 inches Hg back pressure) because, at 5 inches Hg back pressure, expansion of steam to "B" and corresponding work ceases at a higher enthalpy with the result that the increment between point 4 to point 5 is lost to the cycle.

The vertical lines 3 - 4' and 3 - 5' represent the isentropic expansion line for a turbine with 100 percent mechanical efficiency, and the ratio $H_{hr} - H_4$ compared to $H_{hr} - H_{4'}$, or $H_{hr} - H_5$ compared to $H_{hr} - H_{5'}$ is a measure of the mechanical efficiency of the turbine.

The increase in exhaust pressure will result in an increase in the turbine cycle heat rate, since less work is accomplished for the same flow of steam through the turbine. The increase in exhaust pressure will also result in a decrease in the amount of work accomplished by the turbine, assuming constant steam flow.

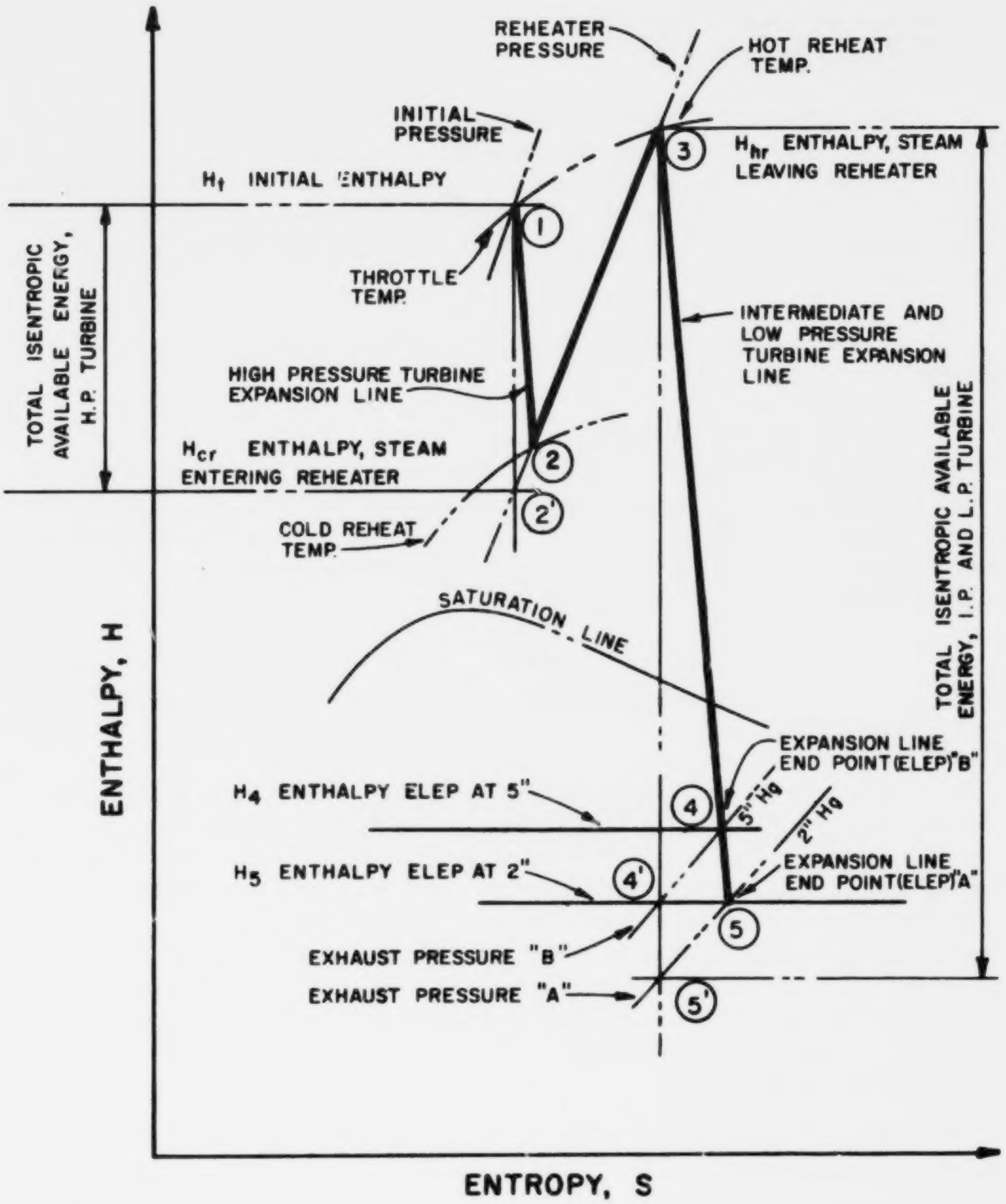

REHEATER PRESSURE
HOT REHEAT TEMP.
INITIAL PRESSURE
H_t INITIAL ENTHALPY
H_hr ENTHALPY, STEAM LEAVING REHEATER
1
2
2'
3
4
4'
5
5'
THROTTLE TEMP.
TOTAL ISENTROPIC AVAILABLE ENERGY, H.P. TURBINE
HIGH PRESSURE TURBINE EXPANSION LINE
INTERMEDIATE AND LOW PRESSURE TURBINE EXPANSION LINE
H_cr ENTHALPY, STEAM ENTERING REHEATER
COLD REHEAT TEMP.
SATURATION LINE
TOTAL ISENTROPIC AVAILABLE ENERGY, I.P. AND L.P. TURBINE
ENTHALPY, H
EXPANSION LINE END POINT(ELEP)"B"
H_4 ENTHALPY ELEP AT 5"
5" Hg
2" Hg
H_5 ENTHALPY ELEP AT 2"
EXPANSION LINE END POINT(ELEP)"A"
EXHAUST PRESSURE "B"
EXHAUST PRESSURE "A"
ENTROPY, S

FIGURE 23 — DIAGRAM OF STEAM EXPANSION LINE

The turbine cycle heat rate may be expressed as a function of heat input and generation.

$$\text{Heat rate} = \frac{\text{heat input}}{\text{kw output}} \qquad [35]$$

Therefore, the change in power output is inversely proportional to the change in heat rate.

Effect of back pressure on heat rejection of turbine. The amount of heat rejected per pound of steam flowing to the condenser is expressed by the equation:

$$Q_r = W_c (H_x - h_c)$$

where: Q_r = heat rejection by exhaust steam, Btu/hr.

W_c = flow of exhaust steam to condenser, lbs./hr.

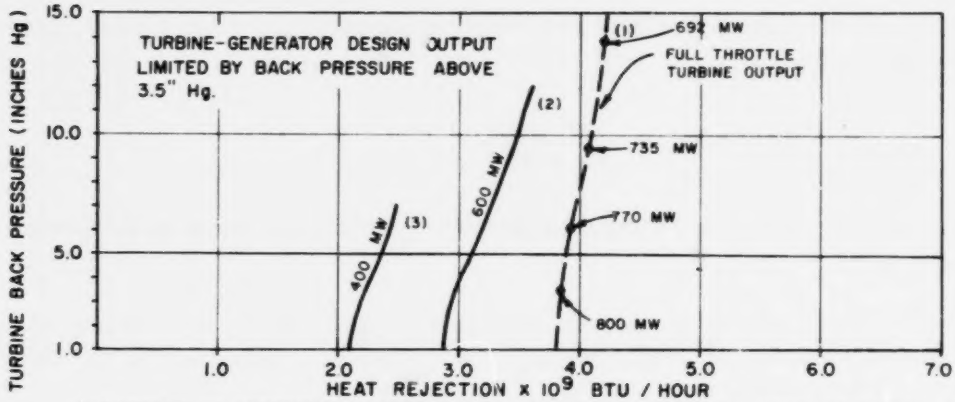
H_x = enthalpy of exhaust steam at expansion line end point, Btu/lb.

h_c = enthalpy of condensate, Btu/lb.

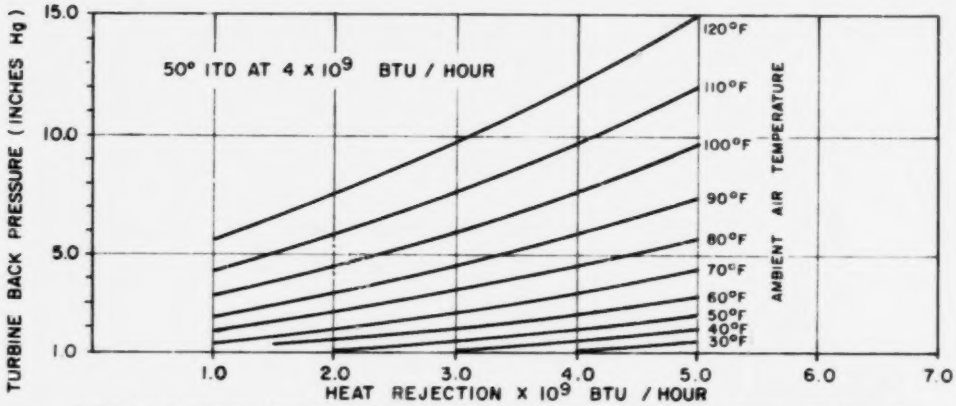
Assuming no subcooling of the condensate, h_c is the enthalpy of saturated liquid corresponding to the exhaust pressure of the turbine. The effect of increased turbine exhaust pressure is increased heat rejection per kwh.

The capability of a turbine manufactured under current standards of design and construction will be reduced for turbine back pressures above 3.5 inches Hg. Figure 24 (a) shows the effect of increased turbine back pressure upon the heat rejection and capability of a nominal 800-mw turbine-generator unit operating at 2,400 psi, 1,000°F/1,000°F throttle conditions. Curve 1, Figure 24 (a), shows the operation of a turbine at full throttle turbine output. Below 3.5 inches Hg back pressure, the capability is greater than 800 mw; at 3.5 inches Hg the capability is 800 mw. At 6.4 inches Hg back pressure, the capability is 770 mw; at 9.3 inches Hg, the capability is 735 mw; and at 14 inches Hg it is 692 mw.

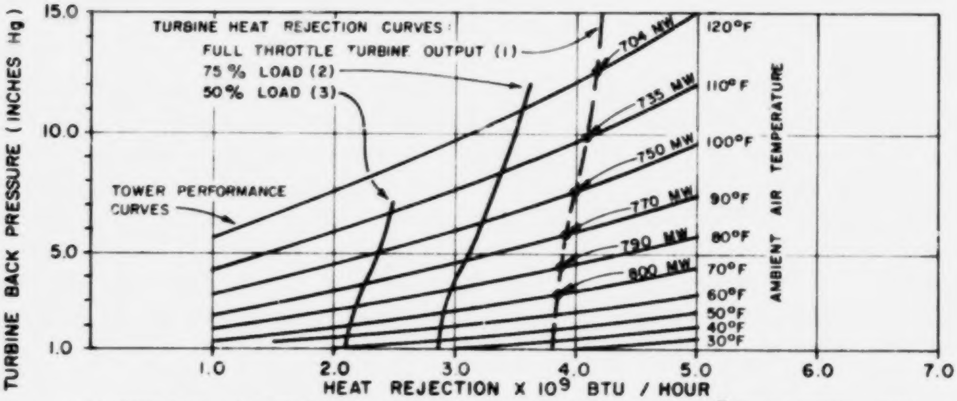
Curve 2 shows the total heat rejection for the same unit when operating at 600 mw, and Curve 3 shows the total heat rejection when operating at 400 mw. Additional throttle flow is required to maintain a constant load when there is an



(a) EFFECT OF INCREASED TURBINE BACK PRESSURE ON TOTAL HEAT REJECTION TO CONDENSER AT VARIOUS TURBINE LOADS FOR AN 800 MW FOSSIL-FIRED PLANT



(b) DRY COOLING TOWER PERFORMANCE CURVES REPLOTTED FROM FIGURE 21



(c) COMBINED PERFORMANCE CURVES OF 50° ITD AT 4 x 10^9 BTU / HOUR DRY COOLING TOWER AND 800 MW FOSSIL-FIRED GENERATING UNIT

FIGURE 24 — DRY-TYPE COOLING TOWER AND TURBINE CURVES

increase in turbine back pressure, assuming the turbine is operating in a load range which will permit the throttle to pass the required amount of steam.

Combining Performance of Cooling Tower and Turbine

Figure 24 (b) is a plot of the heat rejection versus turbine back pressure of the dry-type cooling tower, the performance of which is plotted on Figure 21, revised to show only the performance below 5 billion Btu per hour since this is the range applicable to an 800-mw fossil-fueled generating unit.

If the turbine heat rejection curves from Figure 24 (a) are plotted on Figure 24 (b), as shown on Figure 24 (c), the intersection of the turbine performance curves and the tower performance curves represent the turbine back pressures which will prevail with varying turbine-generator loads and air temperatures.

From the combined curve, it is seen that at 800-mw load and 70°F air temperature, the back pressure will be approximately 3.5 inches Hg. For full throttle flow and 100°F air, the back pressure will be 7.9 inches Hg at approximately 750-mw maximum capability.

Similarly, at 600 mw and 90°F, the back pressure will be 4.7 inches Hg and at 40°F, 1.0 inch Hg.

Comparison of Performance of Dry Tower and Conventional Cooling Systems

Since the variation in annual dry-bulb temperature is greater than that of natural water temperature or water temperature from an evaporative-type cooling tower, the change in turbine back pressure for a generating unit equipped with a dry-type cooling tower will cover a wider range than that of a unit with a surface-type condenser and conventional cooling system.

The wet-bulb temperature of the air is an important parameter in the design and performance of evaporative-type cooling towers since the wet-bulb temperature of the air is the lowest temperature to which the water circulating through the tower can be cooled. The term "approach" is used in evaporative tower terminology to designate the difference between the temperature of the cooled water leaving the cooling tower and the wet-bulb temperature of the ambient air. The design wet-bulb temperature of the air for a specific site is generally selected as that wet-bulb temperature which is exceeded for no more than a small percentage of the time on the average.

The proper selection of the design conditions for an evaporative-type cooling tower for use with a steam-electric generating unit is a complex process and takes into account the capital costs of the tower for various approaches, turbine back pressure variation, pumping and fan power costs and, in general, requires an analysis comparable to the economic selection of a dry-type cooling system.

A wet-type cooling tower with a 15°F approach will cool the circulating water to within 15°F of the ambient air wet-bulb temperature at design heat rejection load. Carrying the design heat rejection load from the condenser, such a tower would cool the water to 100°F when the wet-bulb temperature is 85°F.

Figure 25 shows the variation in average monthly dry-bulb and wet-bulb temperatures for four locations in the United States.

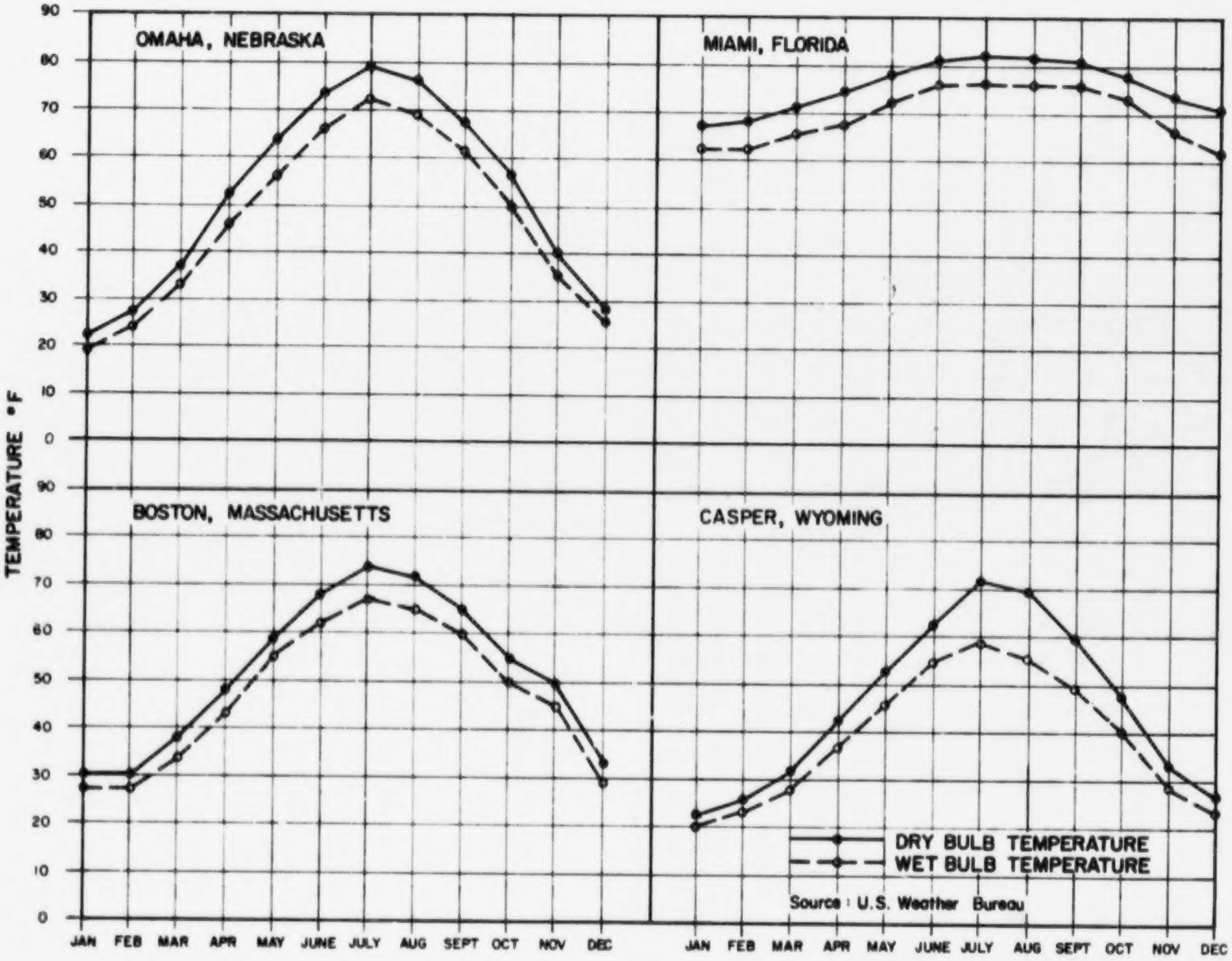
Figure 26 from (9) shows a diagrammatic comparison of the turbine exhaust pressures obtained with typical systems using dry- and evaporative-type cooling towers. Figure 26(a) shows the variation in back pressure and saturated steam temperature corresponding to turbine back pressure under full-load conditions as functions of ambient air temperature for a typical location. Note that for the dry tower (Curve 1), the variation in turbine back pressure has a greater range than for the evaporative tower. This same trend is shown in Figure 26(b) where the variation in turbine back pressure for the two types of cooling systems is shown plotted against varying turbine load with constant ambient air temperature. Curve 2 of Figure 26(a) shows the operating characteristics of a dry tower with a smaller ITD than the dry-type cooling tower shown by Curve 1. Since the turbine cycle efficiency is adversely affected by rise in back pressure, the greater range in turbine back pressure experienced with the dry tower results in a wider range of turbine heat rates as compared to wet tower operation.

Also, greater loss of capability will generally be experienced with units equipped with dry-type cooling towers. Economic studies undertaken in this report indicate that ITD of dry-type towers will be from 55°F to 60°F in areas where average conditions prevail.

In a location typical of the eastern part of the United States, design parameters might be as follows:

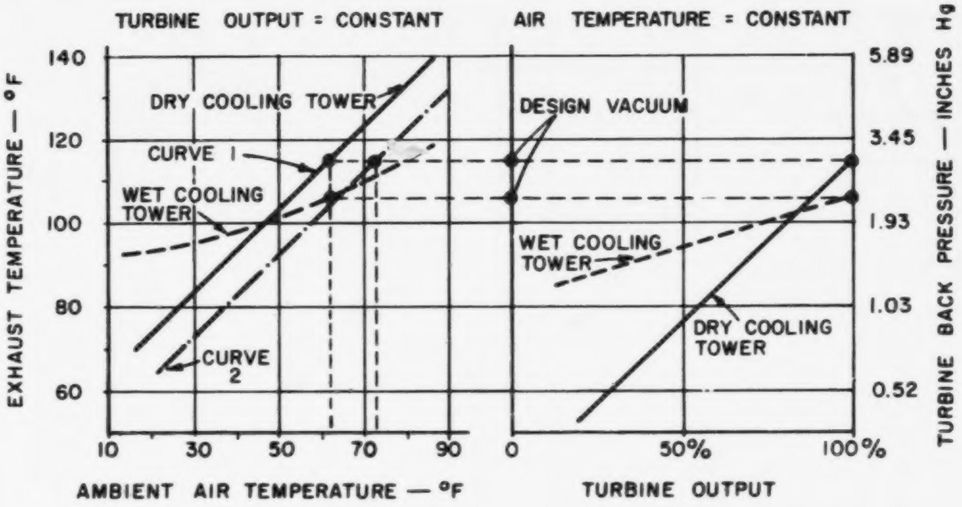
Dry-bulb temperature at 1 percent level:	90°F
Wet-bulb temperature at 1 percent level:	76°F

(Design temperatures at the 1 percent level are the temperatures which are equalled or exceeded by 1 percent of the 2,928 hours of June, July, August and September in an average year.)



Source : U.S. Weather Bureau

FIGURE 25 — TYPICAL AVERAGE MONTHLY TEMPERATURES, DRY AND WET BULB



(a) COMPARISON OF TURBINE BACK PRESSURE AND EXHAUST TEMPERATURE AT CONSTANT LOAD AS A FUNCTION OF AMBIENT AIR TEMPERATURE

(b) COMPARISON OF TURBINE BACK PRESSURE AND EXHAUST TEMPERATURE AT CONSTANT AIR TEMPERATURE AS A FUNCTION OF TURBINE EXHAUST STEAM LOAD

FIGURE 26 — COMPARISON OF DRY TOWER AND EVAPORATIVE TOWER PERFORMANCE (9)

For an evaporative-type cooling tower to serve an 800-mw generating unit, a typical tower might have the following characteristics:

24°F water cooling range

18°F approach to the wet bulb

6°F condenser terminal temperature difference (TTD)

(TTD is the difference between the saturated steam temperature of the turbine exhaust and the temperature of the circulating water leaving the condenser.)

When 76°F wet-bulb temperature is experienced and the turbine-generator is at full throttle steam flow, the condensing temperature for the unit equipped with the evaporative tower is determined in the following manner:

Wet-bulb temperature	76°F
Water cooling range	24°F
Approach to wet-bulb temperature	18°F
Terminal temperture difference	6°F
Condensing temperature:	124°F

For the same generating unit equipped with a dry-type cooling tower of 60°F ITD, the condensing temperature for 1 percent of the summer hours could be expected to be 150°F, determined by adding 60°F to the 90°F dry-bulb design temperature.

The 124°F condensing temperature with the wet-type tower corresponds to 3.8 inches Hg back pressure, as compared to 7.6 inches Hg for 150°F with the dry-type tower.

Thus, for approximately 29 hours per year the unit with the evaporative tower would suffer a 3-mw loss of capability (0.4 percent) while the unit with the dry-type tower would lose 47 mw (5.9 percent).

Since the highest wet-bulb temperatures experienced in the United States at the 1 percent level are approximately 82°F, it can reasonably be expected that loss of capability because of high turbine back pressure during hot weather will not be a major factor with turbine-generator units equipped with evaporative-type cooling towers.

However, there are a number of specific locations in the United States where the dry-bulb temperatures at the 1 percent level exceed 95°F. With 50°F to 60°F ITD, the design range which appears to be a typical economic selection for the United States, the condensing temperatures will be 145°F to 155°F, corresponding to 6.7 inches Hg to 8.6 inches Hg back pressure, with a loss of rated generating capability of approximately 5 to 7 percent (see Table 6, page 88).

The turbine back pressures to be expected with once-through systems are comparable to the back pressures experienced with a typical evaporative-type cooling tower, and, generally, the loss of capability during the summer with either the evaporative-type tower or the once-through system would not be a major factor.

As reported in (29), the highest expected sea-water temperature in Miami is approximately 86°F; in Boston Harbor, 76°F; and in New York City, 78°F. Also, there are few large rivers or lakes which would be considered for a once-through condensing system where the maximum summer water temperature exceeds 85°F.

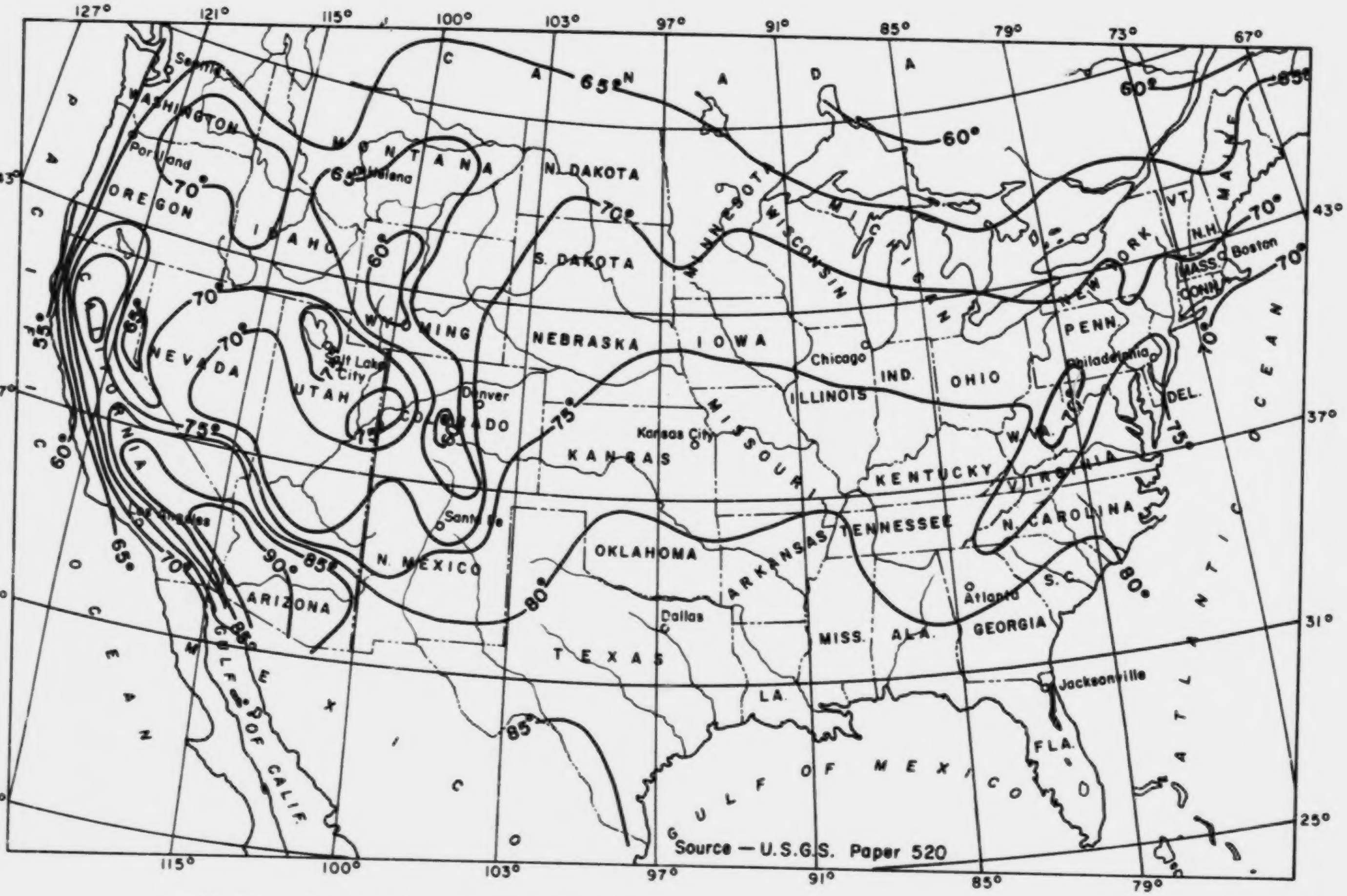
Figure 27, reproduced from USGS Water Supply Paper 520, shows the approximate mean monthly temperature of water from surface sources during the months of July and August for the United States.

It must be emphasized that the 60°F ITD used in the foregoing example for a dry tower, the 24°F cooling water range with 6°F terminal temperature difference, and the 18°F approach for the evaporative tower were selected only for the purpose of illustrating that the increase in turbine back pressure above 3.5 inches Hg and resulting loss of turbine capability are more significant factors with a dry tower than with an evaporative tower. Either type of tower can be selected to have more or less loss in capability if economic considerations justify different design parameters.

Application of Present Large-Turbine Design to Dry-Type Cooling Towers

Available designs. The only design of large turbine-generators presently available from either United States or European manufacturers limits operation to turbine back pressures below 5 inches Hg. Historically, the economics of large utility turbine-generator operations have been such that with conventional cooling systems of the once-through or the evaporative cooling tower type, turbine back pressures have been limited to an upper range of approximately 2.5 to 3 inches Hg. The turbine ratings of presently available turbine-generator units are on the basis of maximum guaranteed kilowatt output at 3.5 inches Hg, with reduced capability for back pressures above 3.5 inches Hg.

According to one leading turbine-generator manufacturer, the experience with high-back-pressure operation is limited to small 3,600-rpm units with short turbine buckets and small exhaust hoods. If the present design of large turbine-



Source — U.S.G.S. Paper 520

FIGURE 27 — APPROXIMATE MEAN MONTHLY TEMPERATURE OF WATER FROM SURFACE SOURCES FOR JULY AND AUGUST

generators were to be used for operation at back pressures above 5 inches Hg, problems would be anticipated in the following areas, unless certain modifications were made.

1. Bucket heating and vibration.
2. Thermal distortion of the exhaust hood and diaphragms which would cause misalignment and rubbing.
3. Abnormal stress caused by thermal cycling.

Possible future designs. There are at least three possible approaches that turbine-generator manufacturers might take to provide a turbine which will operate satisfactorily at back pressures above 5 inches Hg.

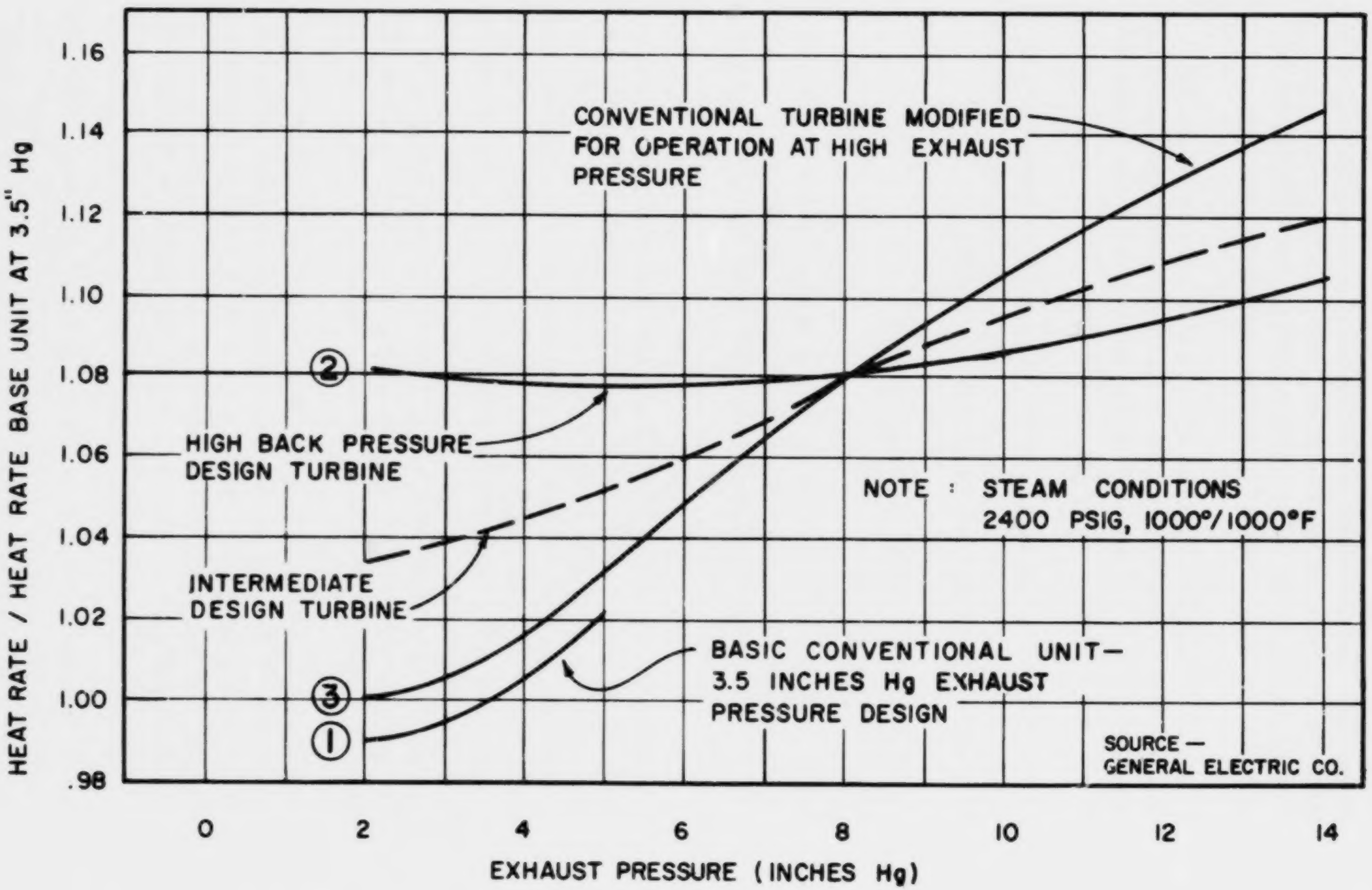
1. Eliminate the Last Row of Blades in the Low-Pressure Turbine of Present Design

   This method has been used by at least one European manufacturer on a 200-mw turbine for use with an indirect-type air-cooling system. The standard 200-mw turbine designed for 2 inches Hg back pressure is modified by removing the last row of blades, 28 inches long, and leaving the next row of 22-inch blades as the last stage to make it possible to operate the unit at higher back pressures. The 200-mw turbine air-cooling system combination is designed for 6.6 inches Hg with 60°F ambient air. With 90°F ambient air, the turbine back pressure will rise to 15.6 inches Hg.

   In addition to the loss of capability which occurs at high back pressure, the capability of the turbine at all back pressures will be less than the capability with the row of 28-inch blades intact as a result of the shortening of the steam path. For this reason, a further modification was made by enlarging the steam flow area of the high-pressure and intermediate-pressure turbines. Also, provision was made to introduce steam into the turbine downstream of the initial stage during times of high ambient air temperature to compensate for loss of capability as a result of high turbine back pressure.

2. Design of a Large Turbine to Operate at High Back Pressure

   One approach to the problem of turbine operation with dry towers is to design a new line of turbines, Curve No. 2 on Figure 28, for operation at back pressures from 2 inches Hg to approximately 15



NOTE : STEAM CONDITIONS 2400 PSIG, 1000°/1000°F

SOURCE — GENERAL ELECTRIC CO.

FIGURE 28 — ESTIMATED TURBINE-GENERATOR, FULL LOAD, HEAT RATE VARIATION WITH ELEVATED EXHAUST PRESSURES

inches Hg. The last-stage blades would be on the order of from 15 inches to 20 inches in length, rather than the 26-inch to 33.5-inch blades used with large utility turbines currently designed for operation up to 3.5 inches Hg back pressure and 3,600 rpm.

The exhaust structure would be considerably redesigned and all stages of the turbine would have to be stronger and pass more steam flow than present designs in order to compensate for the thermodynamic loss associated with high exhaust pressures.

Such a turbine is not now available, nor would we expect any development to be started by manufacturers until there was a demand for a large number of high-back-pressure turbines.

3. Modification of Turbine of Present Design (Curve 3, Figure 28)

Another method available for operation at high back pressure would be the modification of a turbine of present design standards so that it would be suitable for operation at back pressures up to approximately 15 inches Hg.

The high-pressure and intermediate-pressure sections would essentially be the same as for the conventional turbines, except for the changes in steam flow to achieve over-all performance and rating differences, and the low-pressure turbine would have the same total exhaust annulus as for the 3.5-inch Hg design. However, the last several rows of blades would be redesigned for additional structural strength and would be limited to lengths between 25 and 30 inches. The smaller hood structure and shorter bearing span that go with this length of last-stage blade would help to solve the mechanical problems associated with high exhaust pressure.

Figure 28 shows several curves representing the relative heat rates of the different types of turbines described above. The ordinate of the chart shows the ratio of the heat rate of the particular turbine under consideration to the heat rate of the basic turbine at 3.5 inches Hg back pressure operation, represented by Curve No. 1. Note that this curve stops at 5 inches Hg, since 5 inches Hg is the limit of back pressure operation recommended by the manufacturer.

Curve No. 2 represents the relative heat rates for a turbine especially designed for high-back-pressure operation as described in paragraph 2 above. This turbine would have its best performance between 2 inches and 8 inches Hg back pressure, with increasing heat rate above 8 inches. Note that the turbine designed for high back pressure has poorer heat rate performance below 8 inches than the modified turbine of present design, but has better performance above 8 inches. The dashed-line curve represents the heat rate performance of a turbine of high back pressure design with different characteristics than the turbine of Curve No. 2. By tailoring the turbine design to the specific economic considerations for any particular application, it is theoretically possible to have a number of such designs represented with performance between Curves 2 and 3.

Curve No. 3 shows the heat rate performance expected from a conventional turbine modified as described in paragraph 3 above.

Correspondence with another major United States turbine manufacturer indicates that this manufacturer is in the initial stages of a study for high-back-pressure application with dry-type cooling towers and believes that, although it is theoretically possible to modify present turbine designs for high-back-pressure operation, such modification may not be economically or technically feasible with the present state of the art.

The economic evaluation studies in this report were performed on the basis of information furnished by turbine-generator manufacturers for turbine cycle heat rates obtainable with presently designed large turbine-generators modified to operate at back pressures higher than 5 inches Hg, the present limit of back-pressure operation. Although manufacturers are currently studying designs of large, high-back-pressure turbines for operation with dry-type cooling towers, no information as to price or performance is yet available. The economic results obtained in this study may be modified somewhat when turbine-generators designed especially for operation at high back pressure are considered. Both the loss of capacity and the heat rate characteristics of such turbines will be different from the characteristics of conventional turbines. However, cursory studies indicate that the changed characteristics may not significantly alter the production costs as found herein.

Use of Recovery Turbine With Main Circulating Pumps

The circulating water system of the indirect, dry-type cooling tower system is usually designed so that a positive water pressure head of approximately 3 feet at the highest elevation of the cooling coils exists at all times during operation. The purpose of this positive pressure is to prevent air leakage into the coils in case of leaks. Also, with positive water pressure in the coils, any leaks will be apparent to the operators.

In order to maintain positive water pressure in the coils, a restriction of flow must be imposed in the circulating water piping between the cooling tower and the condenser. This restriction could be accomplished by the use of a throttling valve which would be adjusted for varying circulating water flows to maintain the desired pressure at the high point of the coils. However, in order to recover the head that would be lost across a control valve, a water turbine (usually of the Francis design) may be installed in place of a throttling valve. Such a turbine is able to convert approximately 85 to 90 percent of the head drop across the turbine into useful energy and provide from 20 to 40 percent of the power required for the main circulating pumps.

Generally, the recovery turbines are directly connected to the circulating pumps on the same shaft as the pump-driving motor, but the hydraulic turbine could also be used to drive a small generator.

Figure 29 shows a diagram of the pressures in the circulating water system of an indirect dry tower system equipped with a recovery turbine.

Use of Multi-Pressure (Series Connected) Direct-Contact Condensers With Dry-Type Cooling Towers

The large volumes of the steam flow to the low-pressure end of turbine-generators in sizes above approximately 300 mw require that multiple low-pressure turbines and condensers be used. The circulating water flow through the condensers may be either in parallel, with the flow divided for equal volume of flow through the multiple condensers, or the flow may be in series, with the total volume of circulating water flowing through each condenser. Either arrangement of flow can be used with conventional surface condensers or with direct-contact condensers and dry towers.

Parallel circulating water flow through the condensers results in the same pressure in all condensers and also the same final temperature of the circulating water leaving each condenser. With the flow of circulating water through the condensers in series, the pressure in the first condenser will be lower than the pressure in the following condensers as a result of the increase in circulating water temperature entering the following condensers. The total rise in the circulating water temperature will be the same for either parallel flow or series flow.

In the case of series connection of circulating water flow through the condensers, the average pressure in the condensers will be lower than the back pressure obtained with the circulating water flow in parallel through the condensers, assuming the same quantity of exhaust steam and circulating water. Because of the lower average back pressure, there is a slight thermodynamic advantage for series connection of circulating water (also called multi-pressure condensers), and, for this reason, a number of large turbine-generator units in the United States with surface condensers have been constructed with series connection of circulating water.

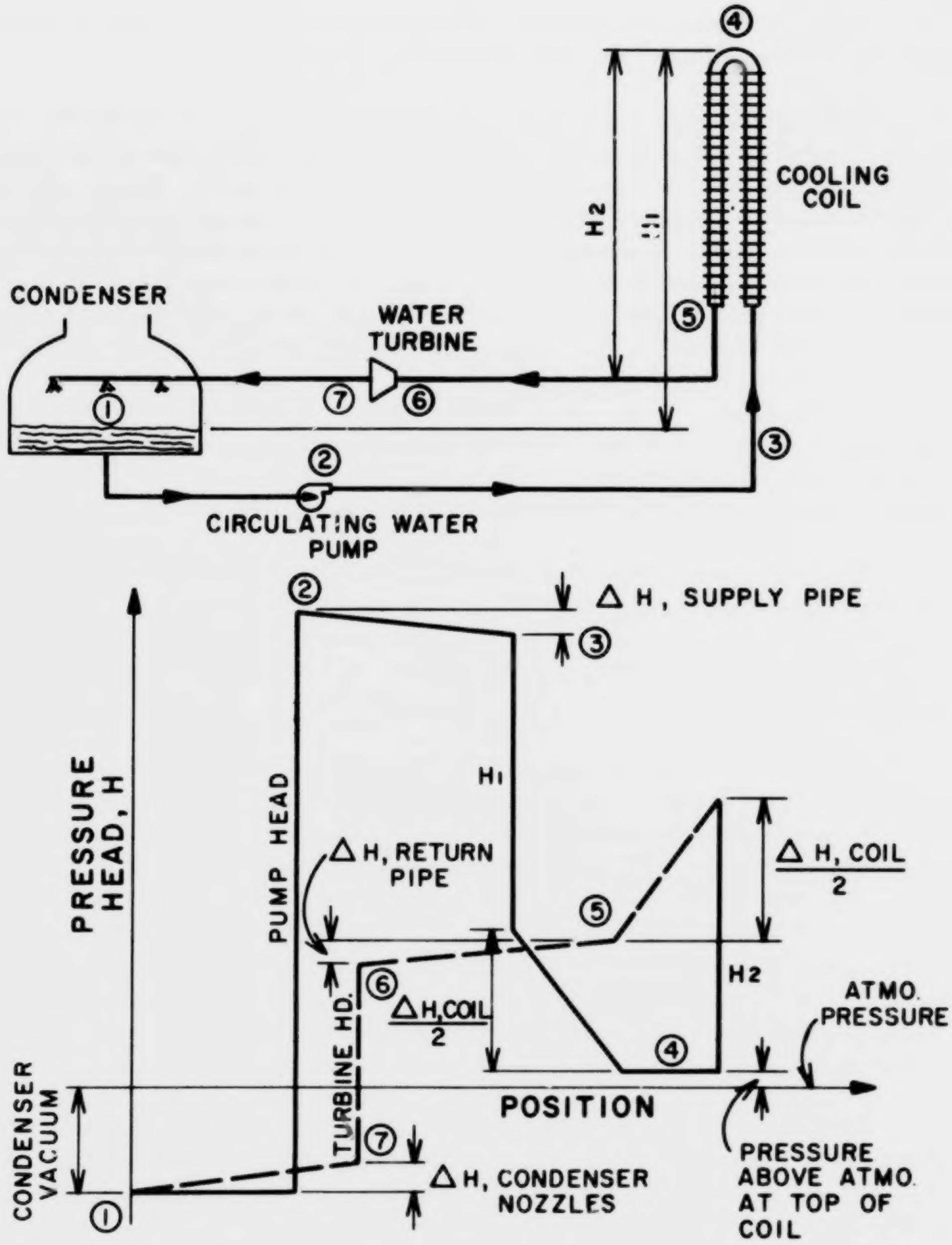


FIGURE 29 — PRESSURE HEAD DIAGRAM FOR CIRCULATING WATER SYSTEM OF INDIRECT DRY TOWER EQUIPPED WITH WATER TURBINE

Figure 30 shows a diagrammatic arrangement of the two types of circulating water connections with surface flow condensers.

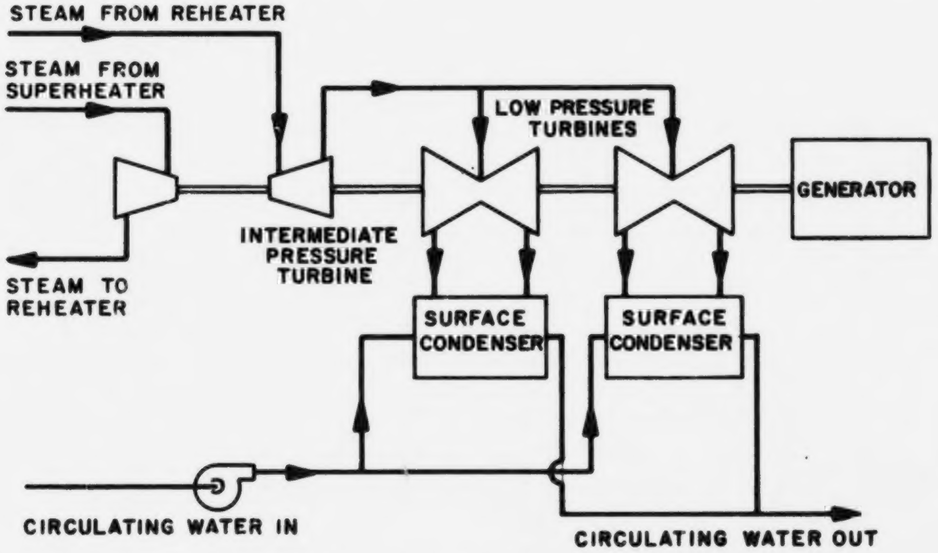
With surface-type condensers, one circulating pump can handle the total flow of circulating water for series-connected condensers since the circulating water flows through the tubes of the first condenser and, subsequently, through the tubes of the following condensers in an integral hydraulic circuit without coming into contact with the steam. However, in the case of the direct-contact type steam condenser used with dry-type cooling towers, the circulating water and steam are intimately mixed in each condenser shell so that it is necessary to convey the mixture of condensed steam and circulating water from the first condenser to each subsequent condenser, either by pumping or by gravity flow. A design has been developed by Dr. Heller which takes advantage of the low pressure drop of the spray nozzles in the direct-contact condenser to permit the transfer of circulating water from one condenser to the next by means of gravity. The downstream condensers are located at a sufficiently lower elevation than the upstream condensers to permit gravity flow (30).

Multi-pressure operation of condensers and series connection of circulating water flow has a distinct advantage with a dry tower installation because multi-pressure operation results in a greater ITD with the same average turbine back pressure and circulating water flow, as compared to the ITD obtained with single-pressure condenser operation and parallel circulating water flow.

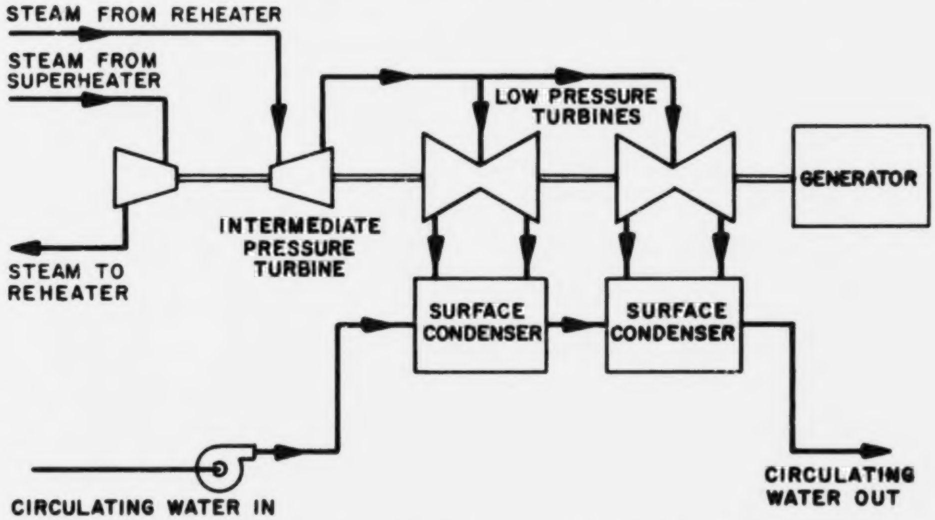
Figure 31 shows the temperature and pressure relations which exist in single-pressure and multi-pressure condenser installations for the same heat rejection and circulating water flow for direct-contact type condensers. In Figure 31 (a), for single-pressure operation T_{w1} is the temperature of circulating water to the condensers, R is the rise in circulating water temperature in the condensers, and T_p is the temperature of saturated steam in the condensers (which is the same as T_{w2}, the circulating water temperature leaving the condenser, assuming no subcooling of condensate). The initial temperature difference for the multi-pressure condensers is ITD_p, and numerically is the difference in degrees Farenheit between T_p and the ambient air temperature.

Figure 31 (b) shows the temperature-pressure relationship which exists in a multi-pressure condensing system with the same heat rejection, the same quantity of circulating water flow, and designed for the same average turbine back pressure as the single-pressure condenser system in Figure 31 (a).

Half of the steam is condensed in Shell No. 1 and half in Shell No. 2. Consequently, one-half of the rise in circulating water temperature occurs in Shell No. 1 and one-half in Shell No. 2. The condenser pressure in Shell No. 1 is the saturated steam pressure corresponding to the temperature of the circulating water leaving Shell No. 1, T_{w4}, which is also the temperature of the circulating water

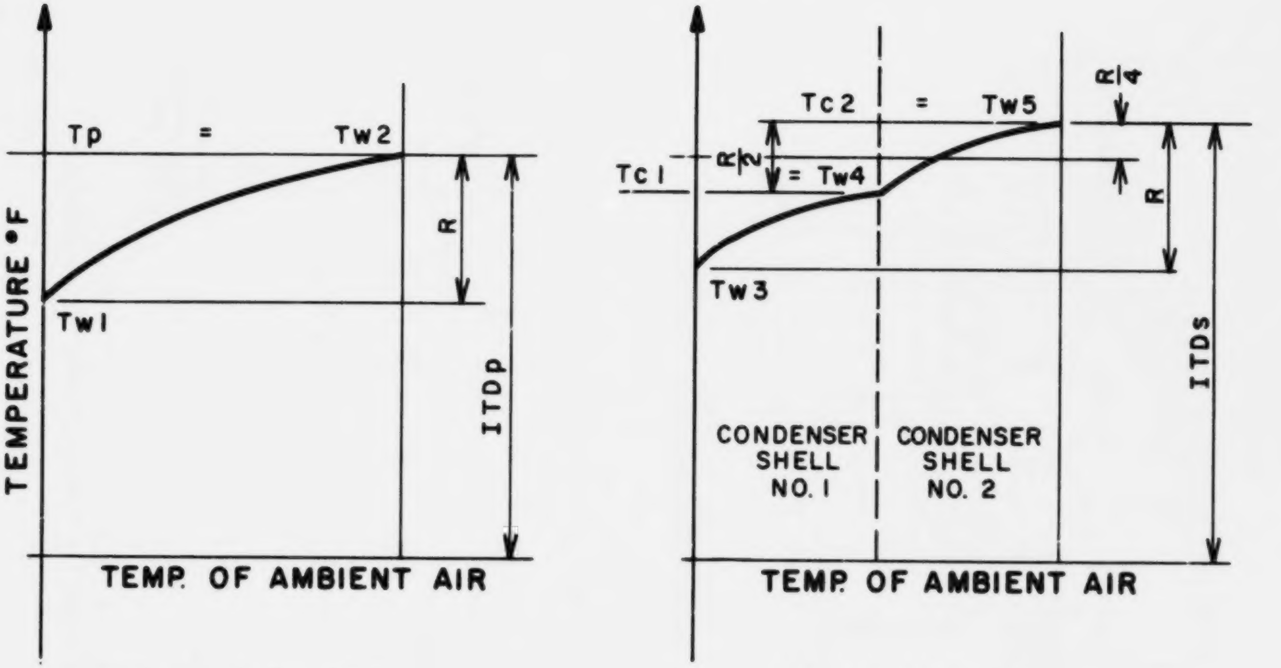


(a) DIAGRAM OF PARALLEL CONNECTION OF CIRCULATING WATER FOR SURFACE CONDENSERS



(b) DIAGRAM OF SERIES CONNECTION OF CIRCULATING WATER FOR SURFACE CONDENSERS

FIGURE 30 — CIRCULATING WATER FOR 4 FLOW EXHAUST TURBINES WITH SURFACE CONDENSERS



PARALLEL FLOW OF CIRCULATING WATER (SINGLE-PRESSURE)
(a)

SERIES FLOW OF CIRCULATING WATER (MULTI-PRESSURE)
(b)

FIGURE 31 — TEMPERATURE-PRESSURE DIAGRAM OF PARALLEL-AND SERIES-CONNECTED, DIRECT-CONTACT CONDENSERS AND DRY COOLING TOWERS (30)

entering Shell No. 2. The condenser pressure in Shell No. 2 is the saturated steam pressure corresponding to the temperature of the circulating water leaving Shell No. 2, T_{w5}. T_{w5} is also the temperature of the water entering the cooling coils of the dry tower, assuming no subcooling.

The average of the condenser pressures of Shell No. 1 and Shell No. 2 for the series connection is equal to the condenser pressure in the parallel flow condenser, but since T_{w5} is greater than T_{w2} by the amount R/4, the ITD of the multi-pressure condensing system is greater than the ITD of the single-pressure condensing system by the quantity R/4.

Since the capital cost of a dry tower is inversely proportional to the ITD, it can be expected that a less expensive dry tower can be constructed for the same turbine back pressure design and circulating water flow rate if the circulating water is connected in series through the condensers.

This conclusion has been verified by actual studies made by Dr. Heller's group for specific plant installations with the result that the estimated capital cost of the dry tower system can be reduced by as much as 10 to 12 percent if the circulating water is connected in series (30).

Effect of Air Temperature at Site

Turbine performance. The variation in ambient dry-bulb air temperature at the site has an effect on dry-type cooling tower performance. The expected air temperatures at any particular location must be taken into account in selection of the ITD of the tower design. Generally, at locations with lower average air temperatures, dry-type cooling towers with greater ITD will be selected than for sites with high average air temperatures.

With any particular tower design, an increase in air temperature will result in higher turbine back pressure and a consequent increase in plant heat rate. When the back pressure exceeds the maximum point at which rated turbine capability can be achieved (3.5 inches Hg for turbines of present design standards), the capability of the turbine-generator is reduced.

Table 6 illustrates the estimated loss of capability at high ambient air temperatures for an 800-mw turbine-generator unit with steam conditions of 2,400 pounds per square inch, 1,000°F/1,000°F, and guaranteed to deliver rated capability at 3.5 inches Hg back pressure when equipped with a dry-type cooling tower of 60°F ITD.

Table 6 shows the reduction in turbine-generator output for back pressures from 1.0 inches Hg to 14 inches Hg back pressure with the corresponding ambient air temperatures for the 60°F ITD tower.

TABLE 6

Variation in 800-Mw Turbine-Generator Capability Due to Changes in Back Pressure With a 60°F ITD Dry-Type Tower

Ambient Air Temperature (°F)	Turbine Back Pressure (in. Hg)	Output (mw)	Percent Rated Capability
19	1.0	809.98	101.2
32	1.5	809.48	101.2
41	2.0	808.98	101.1
49	2.5	806.89	100.9
55	3.0	803.92	100.5
60	3.5	800.00	100.0
65	4.0	795.54	99.4
70	4.5	790.27	98.8
74	5.0	784.03	98.0
77	5.5	777.44	97.2
81	6.0	771.04	96.4
87	7.0	759.53	94.9
92	8.0	748.52	93.6
97	9.0	738.16	92.3
101	10.0	728.25	91.0
106	11.0	718.60	89.8
109	12.0	709.12	88.6
113	13.0	700.34	87.5
116	14.0	692.14	86.5

Table 6 was prepared from performance data furnished by General Electric Company for a tandem-compound, 6-flow turbine-generator modified for operation at high back pressure and is on the basis of full throttle flow performance.

Table 6 does not reflect any possible recovery of capability which might be obtained by taking feedwater heaters out of service, use of over-pressure throttle steam, or by providing a second steam admission point on the turbine with increased boiler capacity.

The 60°F ITD was arbitrarily selected for the table in order to indicate performance for back pressure ranges up to 14 inches Hg with air temperatures which represent a typical site in the United States. Other ITD selections would result in different air temperature-back pressure combinations at full throttle conditions. For example, with a 50°F ITD tower, 3.5 inches Hg back pressure would be obtained with 70°F air rather than at 60°F as shown in Table 6.

Freezing. Air temperatures below 32°F cause potential problems of coil freezing. Provisions must be made in the design of the system to prevent coil freezing during cold weather. The problem of freezing is especially prevalent during periods of light load and during start-up.

The freezing problems which the operators of the existing dry tower plants have experienced and the measures taken to remedy freezing are reported in some detail in Appendix A. It is likely that with a proper automatic control system for start-up operation and shutdown and with adequate alarms, freezing of dry-type cooling towers will not be a problem. However, unless a completely automatic control system for tower operation is provided, much of the success in preventing freezing lies with the plant operators. Thorough training must be given to the operators before a plant is placed into service.

Historically, the freezing of coils which has occurred generally has been during the early period of initial service before the operators were thoroughly familiar with procedures to prevent freezing. A complete automatic control system to manage as many operations as possible is desirable with a dry tower system. Such an automatic control system should initiate and automatically accomplish all functions of taking cooling coils out of service when such action is required because of cold weather and should automatically return the cooling sections to service later on. The control system should also generally perform all tower operations which are necessary to prevent freezing or to operate the tower during freezing weather to the extent that reliance upon the judgement of operators is minimized.

Auxiliary power. With a mechanical-draft tower, more fans will be required during hot weather than during cold weather; consequently, the fan auxiliary power requirements of the tower will be greater during hot weather.

Since the volumetric capacity of the fans to move the required cubic feet per minute of air must be based upon the highest air temperature expected, air temperature variations at the site must be taken into account in selecting the fans and motor drives. The horsepower of the motor driving the fan, however, must be based upon the coldest air temperature expected since the density of the air increases with the lower temperature, while the volume delivered by the fan remains constant for fixed-pitch fans. Variable-pitch fans can be used to reduce fan power requirements at either part loads or during cold weather.

With a tower design which has multiple circulating water pumps, it may be possible to take some of the pumps out of service during cold weather thereby reducing auxiliary power requirements.

Natural-draft cooling tower. The performance of a natural-draft, dry-type cooling tower, with a given ITD and heat rejection load, will be affected by the ambient air temperature in two ways. First, the available draft for moving air through the coils is less at elevated ambient air temperatures than at lower ambient air temperatures. The available draft is reduced, for example, by 14 percent when the ambient air temperature increases from 50°F to 86°F. The second effect is an increase in draft loss through the tower as the ambient air temperature is increased. This effect is caused by the increased volume and corresponding increased velocity of air required to move the same air mass across the coils as compared to operation at the lower temperatures.

As a result of the two effects, the design height of the natural-draft tower must be increased to maintain the required heat rejection performance at higher ambient air temperature locations.

Mechanical-draft cooling tower. Increased ambient air temperatures result in greater air volume requirements for the same mass flow of air (cooling capability requirements). This greater air volume, in turn, results in increased air pressure drop across the heat exchangers. The combination of increased air volume and pressure loss requires increased fan horsepower for mechanical-draft cooling systems operating under higher ambient air temperatures.

Cooling water for auxiliary purposes. The cooling surfaces supplied with standard design of generator cooling, turbine oil cooling, and other auxiliary plant services generally require cooling water of a maximum temperature of 95°F. Because the ambient air temperature will be above the temperature at which 95°F water can be obtained from the dry-type cooling tower during part of the average year, it is necessary to install means of cooling sufficient water for auxiliaries to a maximum temperature of 95°F at all times. The description of the equipment available for this service is found in Section V of this report.

Effect of Precipitation and Humidity

Rain. Based upon the reports of the operations of dry tower plants which have been constructed, rain has an effect upon the performance of dry towers. At the Rugeley Station in England and at the Ibbenbüren Station in Germany, rain results in poorer tower performance. Both of these towers are of the natural-draft type. Rain reduces the draft through the tower because it cools the air inside the tower, consequently reducing the thermal lift. The reduction in draft diminishes the air flow through the cooling coils which causes a higher turbine back pressure. Conversely, the effect of a wetted coil surface is to increase heat rejection performance because of the evaporation of the water on the coil surface. In the case of the large natural-draft, dry-type cooling towers, the small gain in performance from wetted coil surface during rain is nullified by the loss of draft because of the rain cooling the heated air inside the tower shell. A third possible minor effect of rain on the performance of cooling coils is an increase in air-pressure drop through the coils because of the reduction in air passage area caused by water on the coil surface.

Since the flow of air through the cooling coils of a mechanical-draft, dry-type cooling tower is not dependent upon thermal lift, rain does not have an adverse effect upon the performance of mechanical-draft towers. No adverse effects from rain have been reported for the Volkswagen plant or the Neil Simpson plant.

Hail. In areas where hail storms occur, some protection in the form of hail screens should be considered for cooling coils, especially if the coils are installed in a horizontal position. The degree of protection will be influenced by the structural strength of the coil fins and the ability of the fins to withstand distortion or damage from hailstones.

In process industries located in areas prone to hailstorms, it has been customary to use hail screens with forced-draft fans but often not with induced-draft fans, where the fans themselves provide protection for the coil fins.

Sleet or snow. No adverse effects resulting from sleet or snow plugging air passages of the cooling coils of the dry-type tower have been reported.

The Ibbenbüren plant, located in an area where freezing rain occurs regularly during the winter, has not experienced trouble. No problems were encountered at the Neil Simpson plant at Wyodak, Wyoming during heavy snowstorms.

Dr. Heller has advised that plants installed in northern Russia have had no plugging problems caused by snow or sleet. However, louvers on the coil face, or across the area of air inlet, which could afford protection against sleet or snow by closing off the coil sections exposed to the wind, should be considered for natural-draft, dry-type cooling towers to be located in severe-weather zones.

Humidity. Since the temperature of the cooling-coil surfaces is above the dew-point temperature of the air passing through them, the humidity of the air has no noticeable effect upon coil performance. However, fog improves performance of the Rugeley tower, according to published reports (31).

Effect of Wind Velocity and Direction

Natural-draft cooling towers. In general, wind causes poorer performance of a natural-draft, dry-type cooling tower than the performance obtained under conditions of no wind. This adverse effect is generally considered to be due to the low-pressure area which develops on the lee side of the tower as a result of air current eddies caused by the high air flow around the tower. The low pressure on the downwind, or lee side of the tower produces a reduction in static pressure available for producing air flow through that part of the cooling sectors and a higher turbine back pressure is experienced under these conditions.

At Rugeley, no increase in turbine back pressure is experienced until the wind speed reaches 10 mph, but at Ibbenbüren the effect of the wind is felt at lower speeds. The increase in back pressure at Rugeley increases approximately 0.1 inch Hg as compared to approximately 0.3 inch Hg for Ibbenbüren (32) (33).

Dr. Heller has advised that his natural-draft, dry-type towers are designed to maintain guaranteed heat rejection at 4 meters per second wind velocity (9 mph), which is the German standard for the industry for both wet- and dry-type cooling towers. One interesting aspect of the effect of wind upon cooling tower performance in raising turbine back pressure is that wind has an adverse effect upon a natural-draft, wet-type cooling tower as well as upon a natural-draft, dry-type cooling tower. The cooling air flow through the natural-draft, wet-type tower is subject to the same influences of eddies on the lee side of the tower as is the natural-draft, dry-type tower. However, the large mass of cooling water in the wet-type tower storage basin with which the cooled water from the tower mixes before returning to the condenser has a dampening effect upon any immediate influence of the wind on turbine back pressure. With the dry-type tower, the effect of wind is felt immediately since there is no large storage of circulating water.

According to Dr. Heller, a wind velocity of 4 meters per second, for which natural-draft towers are designed, causes a reduction in heat dissipation of approximately 5 percent as compared to calm conditions.

Reports of tests on the Rugeley tower (32) indicate that under high-wind conditions the static pressure on the lee side of the tower was actually higher than the pressure inside the tower and not lower as had been predicted from wind-tunnel tests. These tests also indicate that tower performance is adversely affected by the tangential wind components which tend to reduce air flow through the downstream coolers exposed to the tangential winds. The air flow reduction due to the combination of the above factors more than offsets the beneficial effects of the increased air flow through the coolers on the upwind side of the tower.

M.A.N., a European supplier of natural-draft, dry-type cooling tower systems, uses a design with the cooling sections in a horizontal position inside the base of the tower shell with air flow upward through the coils with a clear space beneath the tube bundles for air passage. M.A.N. has indicated that wind-tunnel tests have shown reduced wind influence on tower performance with this coil arrangement.

A 200-mw installation of such a design is now under construction in South Africa at Grootvlei Power Station and is scheduled for 1972 operation.

Mechanical-draft cooling towers. The influence of average wind velocities upon the performance of dry-type towers equipped with motor-driven fans to move the air through the cooling coils is almost negligible. Operating results of the Volkswagen plant in Germany and the Neil Simpson plant in Wyoming have not indicated any significant influence on tower performance as a result of wind. Both of these plants are equipped with GEA direct-type, air-cooled condensing systems which utilize mechanical draft to move air across the condensing coils.

Effect of Dust

The deposit of dust on the outside surface of cooling-coil fins and tubes has not caused significant difficulties in cooling tower operations. By cleaning the coils periodically with either water or air pressure, the operators of the existing dry-type tower installations have been able to keep the exterior cooling surface sufficiently clean so that performance has not been affected.

The experience with the Rugeley Station tower, however, indicates that local coal dust, which at that station is reported to contain a percentage of chloride compounds, may have been a factor in the severe corrosion which occurred in its cooling coils. The problem was determined to have been corrosion cells set up in the minute cracks between fins and spacer collars of the Forgó coils, likely due to high humidity and atmospheric pollution which resulted in deposits of moisture and chlorides (32). The source of the chlorides has been variously attributed to carry-over from adjacent wet-type cooling towers, the salt-bearing coal dust, and salt-laden fog from the sea coast approximately 150 miles away, but no definite conclusion has been announced.

Although there is a possibility that coal dust was a significant factor in corrosion of the coils at Rugeley, the fact that other dry tower stations have not experienced such corrosion, although coal dust, soot, and dirt have built up on the exterior cooling-coil surfaces, would lead to the conclusion that the Rugeley experience is unique and that utilities considering the use of dry towers could expect little trouble from exterior dirt on the cooling coils.

The plugging of air-cooled coils, as a result of vegetation and debris in the air stream, presents a more serious problem than deterioration of performance from dust and soot, especially at power plant sites in rural areas where material such as cottonwood seed may be present in the air during certain seasons of the year. However, even this problem can be readily overcome by seasonal application of screens and/or vacuum cleaning.

Effect of Radiation and Cloud Cover

Since the cooling surfaces of the dry cooling coils are of such a structural configuration that only a negligible portion of the fin and tube areas are exposed, the effect of radiation from the sun is negligible.

Actual operating experience at Rugeley, as reported by Christopher (31), substantiates the above conclusion; intermittent sun produces only a flicker on the turbine vacuum gauge. However, sunshine and cloudiness are reported to have an influence on the air-cooled, direct-condensing system at the Volkswagen plant.

Effect of Topography

The topography of the plant site utilizing dry-type cooling towers is generally of no great concern in influencing tower performance. The same considerations which govern plant site selection for generating plants with evaporative-type cooling towers or with once-through cooling systems will hold true for dry-type tower sites.

Flat, level terrain is to be preferred. Differences in site elevation may affect the pumping head and auxiliary power requirements of the circulating water system, depending upon the individual design. In general, the site location problems associated with dry-type cooling towers would seem to be of less magnitude than the site problems of wet-type cooling towers, since the dry tower is less susceptible to the problems of recirculation of discharged air (air and water vapor in the case of the wet-type cooling towers), especially when the plant site is located in a valley. Fogging problems as a result of tower discharge are not encountered with dry-type cooling towers.

Effect of Elevation

The elevation of the plant site above sea level must be taken into consideration in the design of dry-type cooling towers. The same considerations which affect dry tower design because of air temperature variations are also factors in the tower design for different locations.

Since a greater volume of air must be moved through the tower at higher elevations to achieve the same mass flow of air, provisions must be made to increase the air flow for towers located at higher elevations. With natural-draft towers, this is accomplished by increasing the height of the tower. For mechanical-draft towers, higher capacity fans must be installed.

According to studies made by Dr. Heller, the capital cost of a natural-draft tower is increased by approximately 4 to 4.5 percent for each 1,000-meter rise in site elevation. Figure 32 shows the relation between required height of a natural-draft, dry-type cooling tower at various elevations and height of the tower at sea level to achieve equal heat rejection performance, as plotted from Dr. Heller's studies.

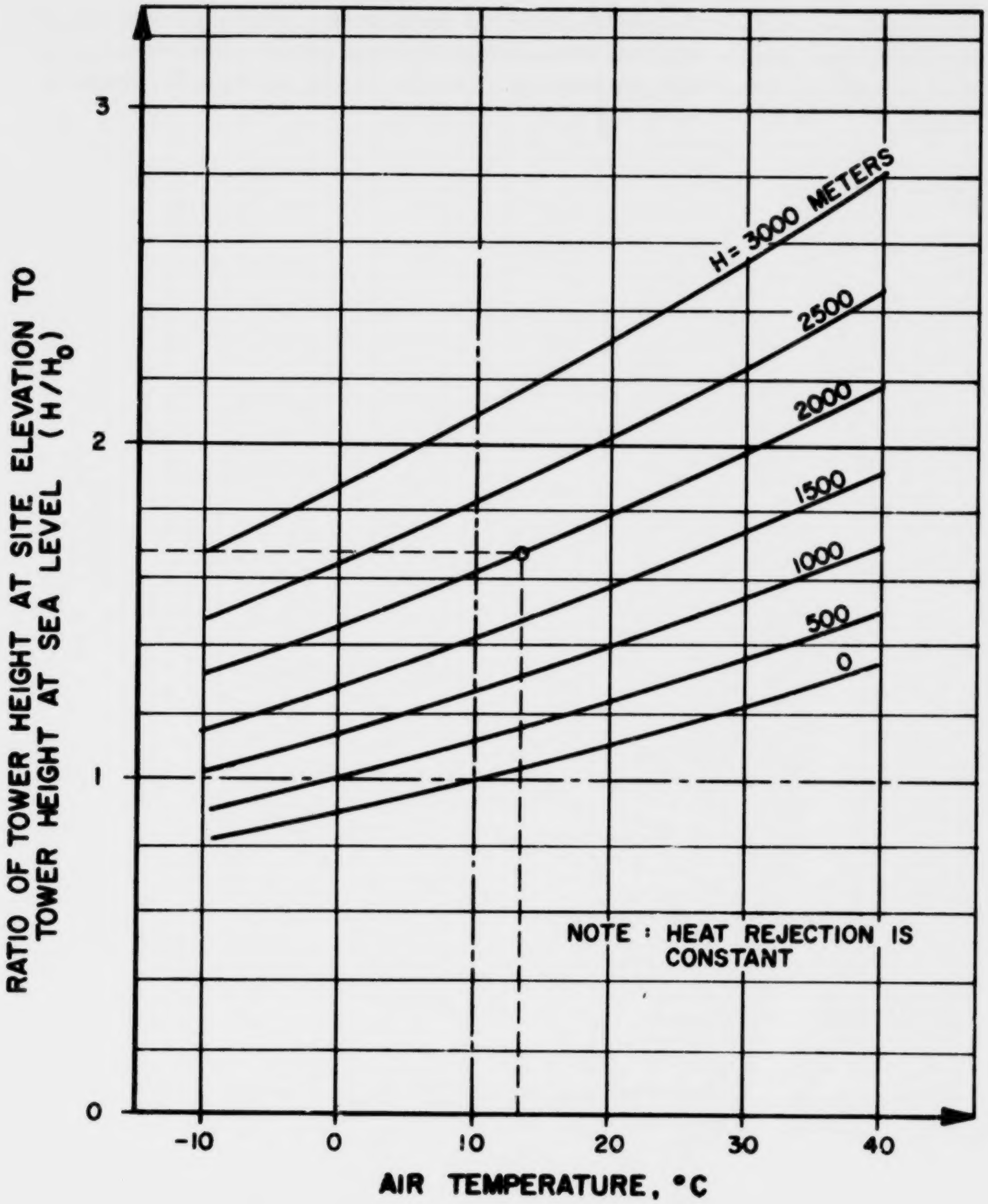


FIGURE 32 — RELATION OF NATURAL-DRAFT DRY-TYPE COOLING TOWER HEIGHT AT VARIOUS ELEVATIONS TO HEIGHT AT SEA LEVEL

SECTION IV

STRUCTURES AND MATERIALS

General

Basically, there are two types of structures used for natural-draft, dry-type cooling towers: reinforced concrete structures and structural steel framework covered with thin siding material.

The type of structure used for mechanical-draft, dry-type cooling towers consists of modular cooling cells of prefabricated components which are assembled at the site. The cells consist of heat exchanger coils, fans, motors and structural steel supports. Generally, the height of mechanical-draft towers is below 100 feet, and the supporting structures are relatively light as compared to natural-draft towers.

The natural-draft tower consists of a shell of either cylindrical or hyperbolic shape, having a height and diameter sized to the air-moving requirements of the particular design. Generally, natural-draft, dry-type cooling towers require less ground area than mechanical draft, dry-type cooling towers of equivalent heat rejection capacity. (See Appendix E for cost data.)

Reinforced Concrete Structure, Natural-Draft Tower

Since the cost of a reinforced-concrete, natural-draft tower of hyperbolic shape is generally less than the cost of an equivalent tower of cylindrical shape—especially in the larger sizes above 400 mw—concrete natural-draft towers are usually hyperbolic in shape.

The hyperbolic concrete tower has a relatively thin concrete shell of varying thickness which is greatest at the base. The shell is terminated at the top of the cooling coils and is supported from the ground by a cross-bracing structure which serves as the supporting columns and also provides the shell opening for air flow. The shell must be stiffened at the top and base with a ring beam to take the concentration of stress at these points. The columns are supported by a continuous ring beam, and piling is provided under each column.

Structural Steel Natural-Draft Towers

The structural steel tower would be of cylindrical shape using prefabricated welded elements for the skeleton and covered with aluminum siding material.

The hyperbolic shape does not seem to be suitable for steel construction because of the difficulty in covering a hyperbolic shell with siding. Also, the hyperbolic shape is difficult to analyze structurally by the membrane theory.

The cylindrical tower would consist of prefabricated structural steel sections to form the main stack of the tower; either bolted or welded into place. The stack must be stiffened at the top, bottom, and at intermediate points by trussed stiffening rings to insure stability and to prevent the stack from becoming oval during wind loading. The main columns should be supported by reinforced concrete pads and piling, since piling is required to counteract the upward force caused by wind loads.

The tower includes a delta roof structure to enclose the additional area required for the base diameter of the cooling coil arrangement.

The steel structure can be either galvanized or painted. A cost comparison of painting versus galvanizing indicates that in the United States the galvanized structure would cost more initially, but would be cheaper throughout the life of the structure, taking into account reduced maintenance and painting costs.

The steel tower would be erected by means of a crane which operates on rings inside the tower, a technique developed by Professor Heller's group. The rings can be left in place as stiffeners.

Design Loadings

The design live loads for steel structures are controlled by wind load on the structure. Seismic loads are not critical because of the relatively light dead load of the steel structure and aluminum siding. The normal wind load is based upon a 100-mph wind velocity at approximately 30 feet above ground level with variation of pressures according to heights. This load should be considered for all areas of the United States except for locations subjected to hurricanes. A wind velocity of 120 mph should be considered for these areas.

The design live loads for concrete structures are generally controlled by wind loads on the structure, except in heavy earthquake areas (Zone 3) as defined by the Uniform Building Code. The wind loads are the same as defined for the steel structures.

Hurricane loads (120 mph wind) develop approximately the same maximum stress condition as for heavy earthquake loadings.

Cost Comparison

Total construction cost of cylindrical steel structures for unit sizes as used in this report will generally be from 15 to 18 percent lower than costs for hyperbolic concrete towers. The cost differential increases as the tower size increases. Also, when tower structures, alone, are considered, it would be considerably cheaper to build one large-capacity tower than two smaller sizes for either steel or concrete. The decisive factor in choosing the type of construction is solely the investment costs for a given locale which, depending on labor rates and material prices, may favor concrete or steel construction.

Tower costs of either material type would be increased from 13 to 15 percent in areas which are subject to hurricanes and heavy earthquake zones.

Corrosion of Coils and Fins

In the design and construction of dry-type cooling towers, particular attention must be given to the possibility of corrosion of the external surfaces of the fins and tubes as a result of atmospheric contaminants, salt-laden fog, or catalytic action between dissimilar metals.

Before selection of the tube and fin material is made, a comprehensive study and survey should be completed at the plant site under consideration in order to obtain information as to the ability of various materials to withstand corrosion.

The Marley Company of Kansas City has recently conducted a series of corrosion and fouling tests to determine which tube and fin materials best withstood exposure at a number of typical power plant sites. To provide accelerated testing, the tube samples were not carrying heated fluid. As a result, the corrosion rate was greater than would be experienced during normal operation of a dry-type cooling tower.

Permission has been obtained from the Marley Company to include the following summary of their test program results:

MARLEY COMPANY

"SUMMARY

CORROSION AND FOULING OF DRITOWER HEAT EXCHANGER SURFACES

> "Corrosion of fins and tubes in the dry cooling tower at Rugeley Station of the Central Electricity Generating Board in England alerted us to the possibility of corrosion and fouling of Dritower heat exchanger surfaces in the USA. To check this possibility, cooperative test programs were established with American Electric Power and Jersey Central Power and Light. The program with Jersey Central Power and Light en-

tailed only external corrosion tests but the program with American Electric Power also included an internal corrosion test.

"The external corrosion test consisted of specially constructed test units in which numerous combinations of alloys and coatings could be simultaneously exposed to a constantly moving air stream. The fins were made of aluminum alloys 1100, 7200, and 3003, and the tubes were of aluminum alloys 3003, 6061, and 5052, and also of carbon steel, copper and admiralty. Some of the samples were electrocoated with acrylic both before and after finning. Five external units were constructed and exposed at sites ranging from sea coastal and heavy industrial to clean rural midwestern. At intervals of 8 and 16 months representative samples were removed, examined, cleaned, and re-examined and the results recorded.

"Internal corrosion tests consisted of sample tube strings of copper, admiralty, and aluminum alloys 5052, 3003, 6061, and welded Alclad 3004. Condensate at 140°F from a supercritical unit was passed through the tube strings at 5 feet per second. At intervals, sections from each string were removed, cleaned, weighed, and the corrosion rates calculated.

"Corrosion of uncoated aluminum fins was severe in the sea coastal and heavy industrial exposures and moderate in the others. There was no significant corrosion of external tube surfaces in any of the exposures. Fouling was slight to moderate except at the sea coastal, heavy industrial and clean rural exposure sites. Vegetation was the sole cause of fouling at the rural midwestern site but corrosion products were an important cause of fouling at the other two. The sections electrocoated after finning showed little corrosion and little fouling at the sea coastal and heavy industrial sites but electrocoating had little effect on fouling at the rural site.

"Internal corrosion rates were significant in all aluminum tubes carrying condensate. Total corrosion was as high as 5 1/2 mills (.0055 inches) for some alloys after 12 months exposure. There was little corrosion of the copper and admiralty tubes in the same period of exposure.

"Significance

"The external corrosion test units were unheated and corrosion effects were greatly accelerated. Therefore, the same effects would not be anticipated in operating units. However, some of the effects could be expected before startup or during periods of shutdown. The extent of corrosion and fouling emphasized the importance of testing potential materials in the expected environment before building the final operating unit.

"The corrosion of aluminum tubes in the internal test was not anticipated. Further work would be needed before aluminum could be considered satisfactory for service in the type condensate used in the test."

Hudson Projects Corporation of Houston, Texas was asked to provide their opinion of the corrosion problems which might be expected with dry-type cooling towers, based upon their extensive experience in the chemical, petroleum and natural gas industries.

Reproduced below is the answer received from Hudson.

HUDSON PRODUCTS

"SERVICE LIFE ALUMINUM FINNED TUBES

"1. Process Industry Air Cooled Heat Exchanger Experience Record

"The air-cooled heat exchange industry has been in existence for nearly 40 years. Industrial air coolers were first used in the gas pipeline industry about 35 years ago as shown below. With time, its applications have grown and continue to grow.

Application	Approximate Year First Placed in Service
Gas Pipelines	1935
Natural Gas Plants	1940
Petroleum Refineries	1945
Chemical Plants	1950

"We have air coolers in service today that are over 25 years old. The aluminum fins have some surface corrosion but the air coolers continue to function and will for many more years. In the last 20 years, we, and our licensees, have manufactured over 600 million square feet of extended surface. Of this amount, we estimate that less than three per cent of these bundles had to be replaced for reasons of corrosion. The replacements were split about equally between internal tube corrosion/erosion and external fin corrosion caused mainly by the presence of acid or halogen gases.

"Our statistical records on air-side fin corrosion are very meager because this has never been a serious problem in the process industry. Of those cases that we are aware, the problem occurred because of a known chemically corrosive gas atmosphere in the area of the air coolers. The towers were comparatively small and installed close to the ground where the corrosive mist was most highly concentrated.

"We could provide you with specifications and a list of our world-wide air cooler installations serving the process industries but we question its value. It would be an impressive list of 'big-name' companies using air cooled heat exchangers in all types of services, but we do not believe it would answer your needs.

"2. Extended Surface Materials and Corrosion Resistance Properties

"In the first 10 to 15 years of industrial air cooler manufacture, copper fins were commonly used as the extended cooling surface. Since then, aluminum has replaced copper because of its large price advantage. The aluminum specification generally used for this fin stock application is shown below.

Type Fins: Aluminum Designation:	Extruded B-241-67 Alloy 6063-0	Plate or Tension Wound B-209-67 Alloy 1100-T-24
Al - %	98.35-97.50	99.0 Minimum
Si + Fe - %	0.55- 0.95	1.0 Maximum
CU - %	0.1	0.2
Mn - %	0.1	0.05
Zh - %	0.1	0.10
Mg - %	0.45- 0.09	-
Cr - %	0.10	-
Ti - %	0.10	-
All Others - %	0.15	0.15 (NMT - .05 ea)

"One of the outstanding features of aluminum is its resistance to corrosion. Aluminum has a great affinity for oxygen with which it combines almost instantaneously to form a protective coating of aluminum oxide. The base aluminum is thus covered by a coating of aluminum oxide which prevents further oxidation and corrosion in the normal air atmosphere.

"Reynolds Metal Company rates the relative cold-test corrosion resistance of aluminum (air-cooler fin-stock material) in various atmospheric surroundings as follows:

Aluminum Designation:	B-241-67 Alloy 6063-0	B-209-67 Alloy 1100-T-24
Rural (Inland areas away from smoke, fumes and industrial dust)	Very Good	Excellent
Industrial (Areas contaminated by smoke, chemical fumes and other industrial dusts)	Good	Very Good
Marine (Areas ranging up to one mile from the sea coast subject to intermittent salt mists)	Good	Good

Reference: "Structural Aluminum Design"; Pages 91 and 113.
Reynolds Metal Company - 1968

"Cold corrosion tests by the aluminum manufacturers (Alcoa and Reynolds) show Alloy 1100 being slightly better than Alloy 6063 due to purity. Plate fin material for power plant service will be B-209-67/Alloy 1100-T-24.

"3. Aluminum Fin Corrosion and Its Prevention

"Aluminum fin corrosion occurs when operating with moisture in a corrosive atmosphere. It is further aggravated when this moisture does not run off but lies on the horizontal fin surfaces allowing the corrosive liquid to work away and destroy the protective aluminum-oxide coating.

"There is little that can be done to clear the air of corrosive gases in some chemical plant installations. However, something can be done about the design and construction of the air cooler and its operation to minimize corrosion.

"Here are our recommendations:

"A. Install induced-draft fans with fan-ring rain-gutters. Induced-draft fans installed on the top of the tube bundle protect the fin surface from direct rain exposure. Fan-ring rain-gutters carry the centrifuged water away from the tube bundles. A forced-draft fan installation, on the other hand, has the entire top face of the bundle exposed to rain water.

"B. Install horizontal tube bundles with the extended fin-surface vertical, or near vertical. Moisture (as a result of dew, mist, fog or rain) will collect and flow off the fin surface rather than lie on it as would be the case with the fins positioned horizontally.

"C. Use variable-pitch, reversible-flow, fans for fluid temperature control. Keep all the tubes in service and warm (10 F to 15 F above ambient air) at all times thereby preventing water condensation on the fin surface. Rather than removing bundles from service at low-loads and low ambient-air temperatures, the air flow can be reduced with variable pitch fans and even reversed with reverse pitch. Reverse pitch will provide co-current flow which ensures warm air at the inlet to the fin tube bundles.

"D. Avoid natural-draft designs which have uncontrolled air flow. Depending upon tower design, ambient air temperature, wind velocity and fluid temperature, some fin surface temperatures can drop below the dew point temperature as a result of air channeling inside the tower structure. This will cause condensation that could promote fin corrosion.

"We believe that the air-cooler corrosion experienced at Rugeley Station was a combination of several factors:

"A. The air-cooler sections were installed vertically around the periphery of the tower with the plate fins in a horizontal position. Moisture from the atmosphere (rain, dew, fog, etc.) and spray from the adjacent wet cooling towers which lay on the fin surface provided the environment for the atmospheric corrosive pollutants. An air-polluting, ash-sintering, plant, which fabricates building blocks from the plant-ash, is built adjacent to the Rugeley Station.

"B. The air-cooler plate fins are joined with compression collars. The resulting tube-to-plate fin joints are not water-tight and hence are subject to corrosion.

"C. The natural-draft hyperbolic tower has no air-control means such as louvers. Under certain operating conditions some of the tube temperatures could fall below the dew point and cause condensation.

"4. Protective Coatings

"Protective coatings on fins such as epoxy, phenolics, etc., may have a place in the process industry but we firmly believe they should not be used in power plant application. We have used, on rare occasions, protective coatings on process-plant air-coolers installed in known corrosive atmospheres. But only the most exceptional power plant locations would ever be subject to such surroundings.

"An industrial-type power plant serving a chemical complex might but certainly not a normal electric utility plant.

"Our principal objections to the use of protective coatings on the extended surface are high cost and degradation of overall heat transfer: All effective corrosion resistant coatings are dielectric in their properties; hence they are inferior heat-transfer materials.

"Fin coating is a poor solution to the problem from an engineering viewpoint because it is preventing the effect and not the cause. A lower-cost installation could be achieved by moving the proposed power plant site away from an existing chemical complex or a sea-coast area. If the power plant stack gases are suspect of being a potential source of the problem, then stack orientation and stack height can be optimized.

"English Electric Company is contemplating the use of an epoxy resin at Rugeley Station but this is to solve a specific problem in a specific situation which they have to live with. We all have learned a great deal since the Rugeley tower was designed about ten years ago.

"5. Simulated Corrosion Tests

"Simulated cold-tube corrosion tests provide a relative corrosion resistance evaluation of alternate materials. It has been our experience that cold-tube corrosion tests are of no value in predicting and evaluating fin service life. A cold metal will generally condense moisture on its surface at least once in every 24-hour period during certain seasons whereas an operating air-cooler may not be exposed to such a moist condition once in a year. It is the moisture which is the catalyst that is operating in conjunction with the corrosive gases that breaks down the protective aluminum oxide film.

"6. Fin Surface Fouling

"We have experienced fin-surface fouling from cottonwood and poplar lint in a few specific installations. When it occurs, it can be vacuum cleaned from the fin surfaces.

"If it is known in advance that the area foliage in the vicinity of the power plant does produce this nuisance, then screens of about 10 mesh size can be installed (during the lint season) below the air cooler bundles. The screens can be cleaned as required, and removed during most of the year.

"7. Power Plant Operation

"Air-cooler fin corrosion in power plants should be less than that experienced in process plants for the following reasons:

"A. With the exception of the boiler stack gases, there should be no corrosive gas producers in the power plant area. Process plants generally have many more potential corrosive gas sources.

"B. Power plant stacks are many times higher than the 40 to 100 ft. stack heights found in process plants. Hence corrosive gases and particulate matter are more effectively dispersed away from the immediate plant area.

"C. Large dry cooling towers serving power plants will have air-coolers installed about 40 ft. (and higher) above grade which is higher than that generally found on the smaller process-plant units. Heavier-than-air chemical gases and pollutants will be of negligible concentration at these heights.

"D. Nuclear power plants presently have a large exclusion area and they produce no corrosive stack gases. Their atmospheric surroundings could be classified as excellent. In the future it is proposed to build nuclear plants close to or in urban areas. From a corrosive gas and particulate release standpoint, future urban areas could also be classified as good or excellent as regards aluminum fin-life expectancy."

Effect of Corrosion on Performance of Coils

Corrosion of cooling coils and fins as a result of atmospheric contamination, salt spray, or other causes would result in loss of heat rejection performance of the dry-type cooling tower if the corrosion were severe enough to change the heat transfer characteristics of the design.

Probably the greatest loss of heat transfer would be suffered if corrosion should destroy the bond between the fins and the tubes so that the path of heat conduction from the tube wall to the fins is broken.

Loss of fin metal by corrosion would reduce the area of heat transfer surface in contact with the air. The products of corrosion, such as metal oxides or sulphides, would have poorer heat conductivity than the pure metal and would impede heat flow. Surface corrosion may also affect the outside film factor and reduce the heat transfer from the fluid inside the tubes to the tube wall because of the poorer conductivity of the metal oxides inside the tube.

Corrosion that resulted in perforation of the tube walls would result in leakage of water in the indirect system and the admission of air in the direct system since the direct-type condensing coils would be at a pressure less than atmospheric. In addition to possibly lowering the heat transfer capability, the air in the coils would result in poorer turbine performance, since it would have the effect of raising the back pressure against which the turbine operates. This is caused by partial pressure of the air in the steam-air mixture.

SECTION V

AUXILIARY EQUIPMENT

General

The principal components of the auxiliary equipment associated with an indirect dry-type cooling tower are:

1. Direct-contact condenser.
2. Circulating water pumps.
3. Water turbines.
4. Air-removal equipment.

The direct-type, air-cooled condensing system does not utilize circulating water pumps or a direct-contact condenser since the exhaust steam from the turbine is conveyed to the cooling coils in the tower and is condensed directly by the air flowing past the coils.

Condensers

In the indirect-type system, condensation of exhaust steam from the turbine is accomplished in the condenser by direct mixing of the circulating water and the exhaust steam. Several designs of direct-contact condensers have been developed by European manufacturers, and at least one United States manufacturer is working on a direct-contact condenser design.

A well-designed condenser must condense the steam with a minimum of subcooling of the condensate below the saturated temperature corresponding to the turbine exhaust pressure, and must also deaerate the condensate and provide for removal of the air and other noncondensable gases. Adequate storage space for the circulating water and condensed steam must be provided in the condenser hotwell. In order to reduce circulating water pumping power requirements, the pressure drop across the spray-water nozzles should be low—in the order of 1.5 psi.

Figure 33 shows a cross-sectional view of the English Electric Company condenser installed at Rugeley Station with a 120-mw unit equipped with a Heller-type dry tower (31).

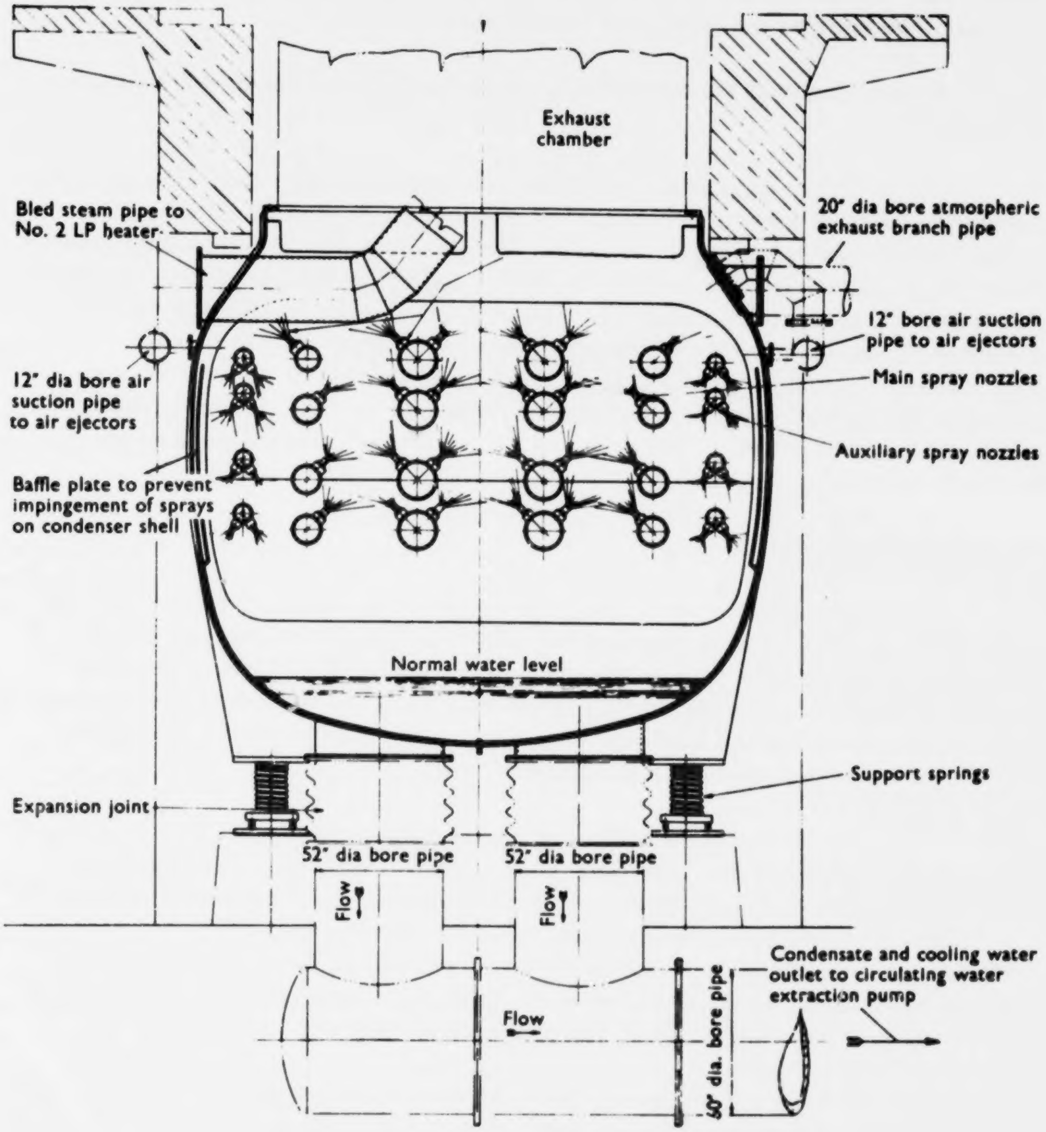

Exhaust chamber
Bled steam pipe to No. 2 LP heater
20" dia bore atmospheric exhaust branch pipe
12" bore air suction pipe to air ejectors
12" dia bore air suction pipe to air ejectors
Main spray nozzles
Auxiliary spray nozzles
Baffle plate to prevent impingement of sprays on condenser shell
Normal water level
Support springs
Expansion joint
52" dia bore pipe
52" dia bore pipe
Flow
Flow
Flow
Condensate and cooling water outlet to circulating water extraction pump
50" dia. bore pipe

FIGURE 33 — CROSS SECTION OF DIRECT- CONTACT CONDENSER USED AT RUGELEY STATION (31)

The Rugeley condenser is a single-shell type, constructed of mild steel. Water boxes at each end supply circulating water to 24 spray-nozzle header pipes running across the shell. The circulating water sprays are directed towards the steam flow, and condensation is achieved as the steam flows downward past the nozzles. The nozzles are constructed of stainless steel.

A direct-contact condenser designed by M.A.N. is shown in cross section in Figure 34.

The M.A.N. condenser is designed for minimum restriction in the exhaust-steam-flow area in order to achieve a low steam-pressure drop from the turbine exhaust flange to the condenser hotwell. Circulating water from the cooling tower enters the condenser at 1 to an annular header and passes through several distribution pipes, 2, to an inner header, 3. The circulating water is sprayed out through a number of nozzle headers, 4, impinging against baffles, 5, and drips downward over a series of plates, 6. The exhaust steam flows downward through the condenser to the hotwell level and turns upward through the cascading circulating water and condensed steam. Air is drawn off across small surface condensers, 7, by water-powered air ejectors, 8.

Air Removal Equipment

For proper operation of a steam condenser, it is necessary to continuously vent off the noncondensable gases and air which are present in the condenser. Also, it is necessary to evacuate the air from the steam space of the condenser prior to putting the condenser into service. The foregoing operating requirements hold true for the direct-contact condenser used with dry-type cooling towers as well as for conventional surface condensers.

Multistage steam-jet ejectors and water-type ejectors have both been used with dry tower installations. Because the air removal capacity of a steam-jet ejector, measured in pounds of air per hour, is reduced when condenser pressure is low, the water-type ejector which uses water as motive power rather than steam is preferred by some European manufacturers of dry tower equipment for the reason that the air-removal capacity of the water-type jet, measured in pounds of air per hour, remains fairly constant over a wide condenser pressure range.

For evacuation of air during start-up, special high-capacity jets which exhaust directly to the atmosphere are used. The start-up jets are shut down when sufficient air has been evacuated for the operating jets, which generally return the steam or water to the cycle, to handle the task.

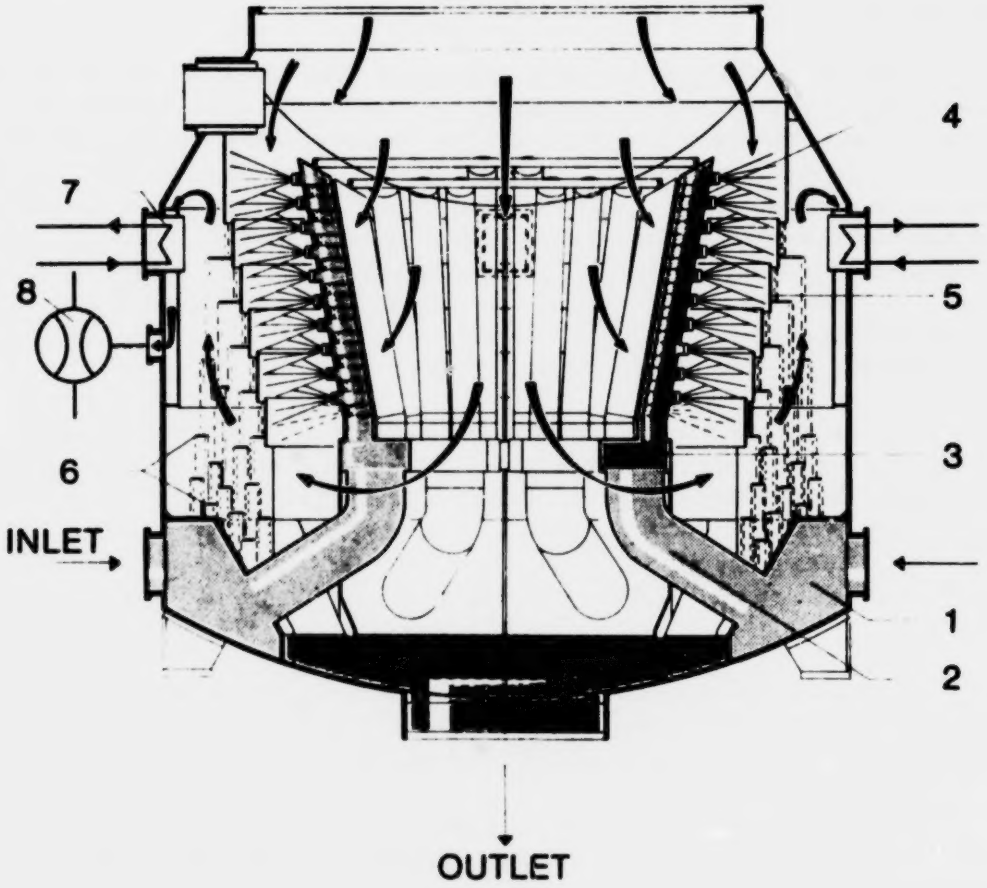


FIGURE 34— M.A.N. DIRECT-CONTACT CONDENSER (9)

Pumps

Since in the indirect-type dry cooling tower system the circulating water from the tower and the exhaust steam from the turbine are mixed together, the circulating pumps must be able to remove water that is at, or very close to, the boiling point corresponding to the condenser pressure. These pumps must have the same design features as conventional condensate pumps; i.e., the suction passages must be generously sized to achieve low velocities because of the high lift against which the pump operates, and adequate provisions must be made to prevent the leakage of air into the parts of the pump under suction pressure.

Because of the large volume of circulating water that is mixed with the condensed steam, the circulating water pumps must handle from approximately 40 to 70 times the amount of water that a conventional condensate pump would handle for the same size unit installed with a surface condenser, depending, of course, upon the temperature rise of the circulating water through the condenser. As an example, an 800-mw turbine-generator equipped with a conventional surface-type condenser requires removal of approximately 7,400 gpm of condensate from the condenser hotwell. The same size unit with an indirect-type dry cooling tower and direct-contact condenser and designed for a 30°F rise in circulating water temperature requires that 300,000 gpm of water be removed from the hotwell.

Although such large pumps designed for removing water from a chamber under high vacuum have not been constructed in the United States, the technology and design experience is readily available. Circulating water pumps for large, indirect, dry-type systems will, in effect, be conventional circulating water pumps, either of split-case horizontal configuration, or of the vertical type, modified for operation with high suction lift. The pump head would be approximately 80 to 100 feet.

Since the direct-type, air-cooled condensing systems do not require circulating water pumps, the condensate pumps used with a direct system would be similar to the condensate pumps used with conventional surface condensers.

In order to reduce the plant water make-up to a minimum, we would expect that the circulating and condensate pumps used with dry-type towers would utilize mechanical seals in place of shaft packing.

Recovery Turbines

For large generating units equipped with dry-type cooling systems, it is economical to recover the excess head imposed on the tower to maintain positive pressure on the cooling coils. For this purpose, recovery turbines would be installed in the circulating water piping between the tower and the direct-contact condenser. Approximately 20 to 40 percent of the total power required to pump the circulating water through the system can be recovered.

The recovery turbines would be conventional, low-head, hydraulic turbines, probably of the Francis type, and would be similar to the turbines of current design and manufacture used with hydroelectric generators.

For the 800-mw fossil-fueled unit considered in this study, the total capability of the recovery turbines would be approximately 3,000 horsepower, operating at a head of approximately 30 to 40 feet.

Power from the recovery turbines could be utilized by either of two methods:

1. Direct connection to the shaft of the motor-driven circulating water pumps; or,

2. Connection to a generator which would produce electrical power for certain auxiliaries.

Auxiliary Cooling

The coolers of the generators, turbines, and auxiliary equipment of a steam-electric generating plant are generally designed for use with cooling water having a maximum incoming temperature of 95°F. Since the temperature of the circulating water from a dry-type cooling tower will exceed 95°F during much of the year, depending upon the ITD selected for the tower, it is necessary to provide auxiliary cooling water from a source other than the main tower. For an 800-mw unit, the auxiliary cooling is approximately 50 million Btu per hour—approximately 1.3 percent of the condenser heat rejection requirement.

There are several methods by which cooling water of an appropriate temperature could be provided for cooling generators and auxiliaries. It would be necessary to make an economic evaluation of these different systems, described below, for each particular plant before determining which would be the proper selection.

1. A small wet-type tower could be used. Because the temperature of cooling water circulated through a wet-type cooling tower approaches the wet-bulb temperature of the ambient air and since the wet-bulb temperatures would not exceed approximately 85°F at any location in the United States, it is possible to obtain 95°F cooling water during hot weather.

2. An evaporative spray for cooling ambient air could be used in connection with a small mechanical-draft, dry-type tower sized to handle the auxiliary heat rejection load. The air to the cooling coils would be cooled by the evaporation of the spray water. Since it is possible to cool the air to within a few degrees of the wet-bulb temperature of the ambient air, an auxiliary cooling-water supply of 95°F could be obtained with an air cooling coil of low ITD design.

3. A third method of supplying 95°F cooling water for plant auxiliaries would be to use a small mechanical-draft, dry-type cooling tower which would operate on the wetted-fin principle. The surface of the cooling coils would be wetted with water as the air passes over the coils. With proper selection of the ITD of the cooling coils, it would be possible to obtain 95°F auxiliary cooling water. Special provisions would have to be made to clean the coils of scale which might accumulate.

4. Mechanical refrigeration could be used to cool a portion of the main circulating water supply to 95°F for auxiliary cooling purposes during periods when the main supply exceeds 95°F. Standard water-chilling equipment could be adopted for this use.

5. In 1958 at the World Power Conference, Professor Heller presented a method of providing cooling water for auxiliaries during periods when the dry tower circulating water exceeded temperature limits suitable for plant auxiliary cooling by using steam-jet refrigeration (34). The use of steam-jet refrigeration has been well-established in the refrigeration and air-conditioning industry and, although this method is not commonly employed, steam-jet refrigerating systems have been used for over 50 years—especially in certain process industries.

Figure 35 shows a schematic diagram of the cycle presented by Professor Heller as it would apply to a dry-type tower plant for providing auxiliary cooling water during periods of high ambient air temperature. Condensate-purity circulating water is pumped through the auxiliary cooling system and sprayed into the evaporator where a small portion of the water flashes into steam because of the low absolute pressure maintained in the evaporator by the steam-jet ejector. Approximately 1 percent of the auxiliary cooling water is flashed to steam for every 10°F

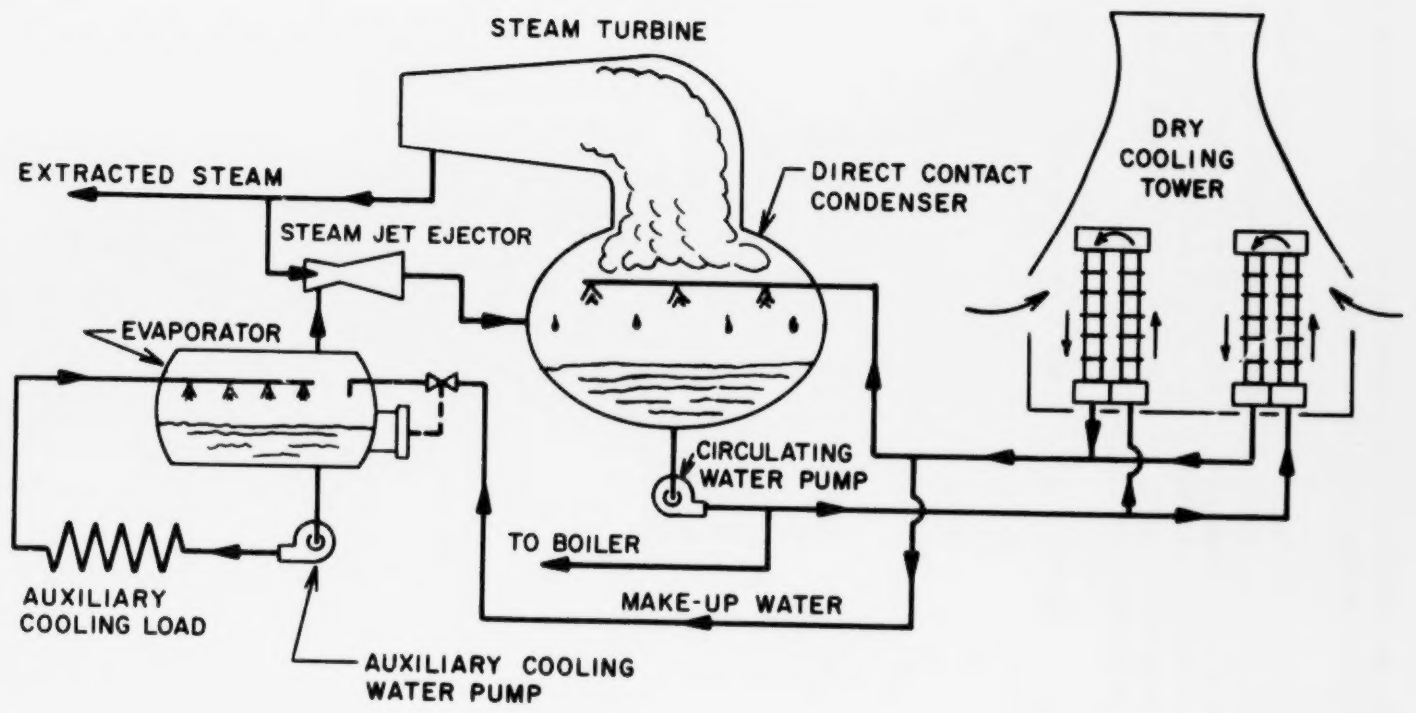


FIGURE 35 — AUXILIARY COOLING BY STEAM-JET REFRIGERATION

of temperature drop in the evaporator. The pressure in the evaporator would be held at 1.66 inches Hg to maintain a temperature of 95°F.

Motive steam to operate the jet would be obtained from steam extracted from the turbine. The steam-jet ejector maintains the required low absolute pressure in the evaporator and discharges the mixture of motive steam and flashed water to the main condenser where it is condensed and pumped to the cooling tower along with the main circulating water supply. The flashed auxiliary cooling water is replaced from the cooled circulating water supply.

Since the system is closed, no water is lost to the cycle with this type of refrigeration system.

SECTION VI

DRY-TYPE COOLING TOWER USE WITH BINARY CYCLES

General

The use of two working fluids having different temperature-pressure characteristics in a power plant cycle is identified by the term "binary cycle". In the binary cycle, a fluid of relatively low vapor pressure is used in the higher temperature (top) section of the cycle and a fluid of high vapor pressure is used in the lower temperature (bottom) section. A number of binary cycles using various fluids, usually with steam as the bottom fluid, have been proposed and a small plant using the mercury-steam binary cycle was constructed in 1930 by the Hartford Electric Light Company. Other fluids investigated as the top fluid in a binary cycle are diphenyl, diphenyloxide, aluminum bromide and zinc ammonium chloride.

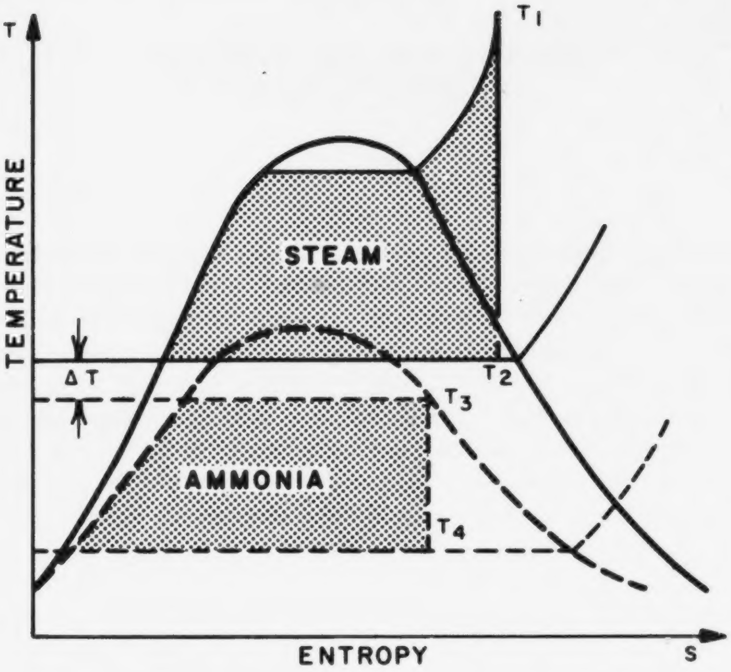
Slusarek (35) has presented a study of the economic feasibility of a binary cycle using steam as the top fluid and ammonia as the bottom fluid, which has particular appeal for use with a dry-type cooling tower. Other studies of binary cycles with dry-type cooling towers using commercial refrigerants as the low temperature fluid, are currently underway by European manufacturers.

Description of Steam-Ammonia Binary Cycle

Figure 36, from (35), shows the temperature-entropy (T-S) relationship and the basic flow diagram of the steam-ammonia binary cycle. In the upper part of the T-S diagram, steam is the working medium and is expanded through the steam turbine from temperature T_1 to T_2, flowing into the steam-to-ammonia heat exchanger where the steam is condensed as it boils the ammonia, which is the working fluid in the lower part of the cycle. The temperature difference, Δt, is necessary to transfer heat from the condensing steam to the boiling ammonia in the heat exchanger. The ammonia vapor at temperature T_3 expands through the ammonia turbine to temperature T_4 where it is condensed and flows to the heat exchanger for recycling.

Conclusions

There are a number of theoretical advantages in the use of the steam-ammonia cycle. The ammonia turbine is much smaller than a low-pressure steam turbine which would be required for a condensing steam cycle since the specific volume of ammonia vapor is lower than steam at corresponding temperature. By terminating the steam cycle at a pressure above atmospheric (34.8 psia in the study



BINARY CYCLE TEMPERATURE—ENTROPY DIAGRAM

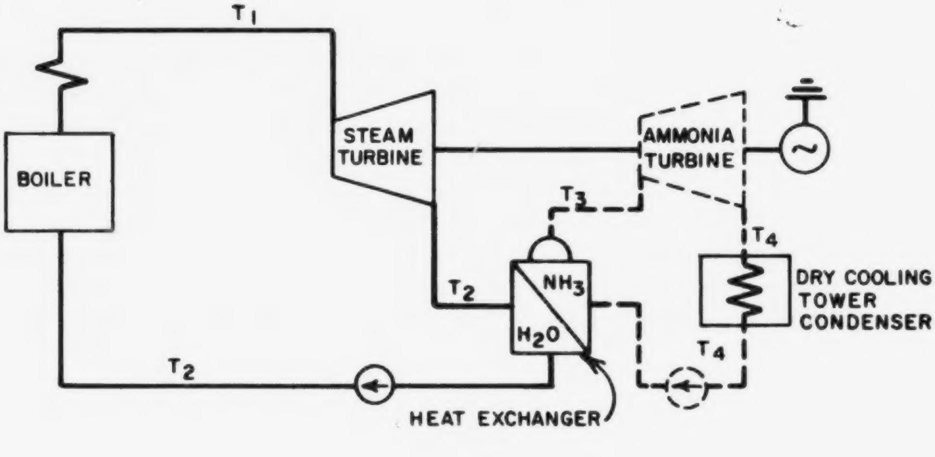


FIGURE 36 — FLOW DIAGRAM OF BINARY CYCLE WITH DRY COOLING TOWER

by Slusarek), vacuum pressures lower than atmospheric are not used in the cycle, because the saturated pressure of ammonia at the lowest probable ambient temperature is above atmospheric. The low specific volume of the ammonia vapor exhausted from the ammonia turbine would permit direct condensation in a dry-type cooling tower with smaller equipment. Thus there is opportunity for reduced cost of a smaller ammonia turbine plant to offset the added cost of a dry-type cooling process. Also, the temperature at which ammonia will freeze (−103°F) is so low that there is no danger of freezing in an air-cooled condensing plant.

The efficiency of a dry ammonia turbine stage is given as 85 percent (35). Mechanical losses, stage losses, moisture losses and exit losses must also be subtracted from the stage efficiency. The resulting ammonia stage operating efficiency is in the order of 74 percent for design conditions and reduces further for off-design performance conditions. A further reduction occurs in the temperature gap in the isothermal heat exchanger where the condensing steam gives up heat to evaporate ammonia, the vapor of which is used to drive the ammonia turbine. The extent of this temperature gap determines this loss of efficiency which amounts to approximately 1 percent per 7.7°F of temperature gap (35). The temperature gap can be reduced by a more expensive heat exchanger, but an optimum must be selected which considers the higher cost of heat exchanger versus higher fuel consumption.

A total steam-ammonia binary cycle generating plant promises an over-all plant efficiency of approximately 42 percent with further improvements to result from feedwater regenerative heating which was not included in the analysis by Slusarek (35). In addition to making available a somewhat higher plant efficiency than is normally obtained with a standard steam plant using once-through or evaporative-type cooling, the binary cycle uses a dry cooling tower which allows mine-mouth plant location in arid areas.

aircraft runway. Dry-type cooling towers would be even more effective in fog dissipation since the heat discharge from the dry-type cooling tower will be greater and jet engine exhaust contains a significant amount of water vapor.

Precipitation. It is doubtful that the precipitation pattern of the surrounding area will be altered by the operation of a dry-type cooling tower. There may be a slight decrease in precipitation in the immediate vicinity of the cooling tower, but it would probably not be measurable.

Air currents. Fritchen et al (38) found in their study of the meteorological effects of a forest fire with a heat release of the order of magnitude of that of a large dry-type cooling tower that the air currents in the immediate vicinity of the fire were significantly altered.

In the case of the dry-type cooling tower, a convergence zone would be formed over the tower which would redirect and alter the speeds of local winds. This effect would vary with local wind conditions and be most pronounced with low wind speeds.

The strength of the updraft will also depend on the local micro-meteorology and will be specifically related to stability, wind speed, and ambient air temperature. Because of the turbulence encountered in strong updrafts, the area should be avoided by aircraft.

Dry-Type Cooling Towers and Air Pollution

If there is an effluent discharge from the power plant associated with a dry-type cooling tower, the best place to vent the effluent would be in the updraft from the cooling tower. This would carry the pollutants up into zones of higher winds where the particulates and gases would be greatly diluted and dispersed through a larger volume of air. This redistribution of the pollutant load would be beneficial locally, but will still add the same amount of contamination to the total pollution problem.

Under certain meteorological conditions, the updraft may break through an inversion and disperse the pollutants above a layer they may have otherwise been trapped beneath. This also would be a beneficial local effect.

How large an area surrounding the plant will be vented by the updraft depends upon the local micro-meteorological parameters and the location and emission factors associated with other sources.

Comparative Effects of Various Cooling Methods

Once-through cooling. The most undesirable side effect of once-through cooling in some installations is the heat addition to natural waters and possible damage to the marine environment of the recipient body of water. The extent of such damage will depend upon the size and mixing properties of the lake, river or ocean into which the warm water is being released.

In addition to the possible environmental damage to the marine life, this method of cooling also produces large amounts of water vapor. Depending on local climatic conditions, this can become a source of fog and mist downwind and cause serious icing problems on adjacent towers and transmission lines.

Cooling ponds. Cooling ponds are another source of large amounts of water vapor and will produce the same undesirable side effects associated with water vapor described for once-through cooling.

Wet (evaporative) type cooling towers. In addition to the problems associated with water vapor production, the wet-type cooling towers add to the air pollution problem through drift losses. Waselkow (39) also experienced main transmission line flash-overs due to cooling tower drift losses.

If production of SO_2 is associated with a wet-type cooling tower, the mixing of the two effluents will cause a major pollution problem. The rate of oxidation of SO_2 to sulphuric acid is enhanced by increased relative humidity. According to Aynsley (36), the rate of oxidation increases rapidly when the relative humidity reaches 80 percent. Thus, a release of SO_2 into the effluent from a wet-type cooling tower can produce deleterious results.

Natural-draft versus mechanical-draft towers. Pollution concentrations and temperature increases will be lower with natural-draft than with mechanical-draft towers. The reason for this is that the updraft from hyperbolic natural-draft towers persists longer and go higher than plumes from mechanical-draft towers. Thus, pollutants and temperature changes will be diluted in larger volumes of air.

Underlying causes of the higher plume, according to Aynsley (36) are the following:

1. The release area of natural-draft towers is higher than that of mechanical-draft towers.

2. The natural-draft tower is constructed in a manner which complements the natural flow.

3. In mechanical-draft towers, fans produce small scale turbulence which tends to break up the plume.

The original speed of ventilation with natural-draft towers is not as great as with mechanical-draft towers, but the speed is more quickly dissipated from the mechanical eddies formed by the fans.

Conclusions

Before the meteorological effects of a dry-type cooling tower can be predicted for any given set of conditions, a thorough model of the meteorological implications of such system must be developed through analysis of actual observations. It is hoped that the first available tower will be used for a complete pilot study. Measurements of temperature (using both horizontal and vertical grids), wind speed and direction, and humidity should be taken in and around dry-type cooling tower sites before and after plant start-up. Efforts should also be made to collect long-term meteorological data from the area to determine if any changes in weather patterns can be identified which are related to the operation of the dry-type cooling tower.

It is our general conclusion that the release of heat into the atmosphere from a dry-type cooling tower will be much less harmful to the environment than the combined release of heat and water vapor associated with other cooling methods. In addition, harmful effluents would be effectively dispersed by inclusion into the updraft from a dry-type cooling tower, whereas the combination of certain pollutants with wet plumes would compound rather than alleviate the pollution problem.

While it is our opinion that a dry-type cooling tower will not produce a measurable effect on a region's climatology, the worldwide buildup of thermal releases and the resultant climatological effects remain important considerations. Thus, on either a local or global scale the subject of the meteorological effects of releasing large amounts of heat into the atmosphere raises many unanswered questions and should be investigated extensively in the next few decades.

SECTION VIII

DISCUSSION WITH MANUFACTURERS

Introduction

During the course of preparation of this report, the authors met with manufacturers and engineers who are currently engaged in dry-type cooling tower work, either in production of components or in development engineering, and also with manufacturers who, although not engaged in actual production of dry tower components, are conducting research and studies on the matter.

In order to obtain first-hand information as to the basic principles, construction and operation costs, operational history and current state of the art of dry towers with steam-electric power plants, conferences were held with various individuals and representatives of manufacturers as hereinafter described.

Dr. Lászlò Heller and Hőterv

Dr. Lászlò Heller, of Budapest, Hungary, serves as the Head of the Department of Energetics of the University of Budapest and as Technical Director of Hőterv—a 1,200-man engineering firm charged with the development of the dry-type cooling tower and the design of industrial plants in Hungary. Dr. Heller presented the initial concept of the indirect dry-type cooling tower system at the World Power Conference in 1956, and, subsequently, has been responsible for the design either in toto or as a special consultant for the Heller-type cooling towers designed to date. Dr. L. Forgò, who serves as assistant to Dr. Heller, developed the cooling coil used with the Heller system. The marketing of the dry tower system components manufactured is under the direction of the Hungarian firm Transelektro, which is also responsible for the manufacture and sale of all electrical equipment.

Hőterv has designed a series of cooling towers in sizes up to 900 mw and conducts computerized studies for manufacturers and utilities throughout the world.

Dr. Heller furnished basic performance data of natural-draft, dry-type Heller towers to the authors. These data were very helpful in the development of the computer program used in the determination of the optimum ITD for various geographical locations in the United States.

An interesting development by Hőterv is a natural-draft dry tower with a steel structural frame and an aluminum skin. The tower is cylindrical in shape, rather than hyperbolic, and is estimated to be somewhat less expensive to construct than the reinforced concrete hyperbolic towers. A special erection technique has been developed by Hőterv utilizing steel rings inside the tower on which a construction crane operates to hoist pieces of the structure. The rings are raised by hydraulic jacks and certain rings are left in place permanently for stiffening the structure. After erection of the tower has been completed, the crane and erection ring are lowered by helicopter.

During the author's trip to Hungary, a visit was made to the factory where the Forgó coils are manufactured. The factory is located near the Town of Jaszbereny and is a part of the Hungarian government's program to bring industry to rural areas. Besides the Forgó coils, home refrigerators and truck radiators are produced at the factory.

Aluminum tubes and aluminum strips for the fins are received from another factory and constitute the raw material for manufacturing the Forgó coil. Much of the labor of putting the coils together is done by hand, in keeping with the program of providing jobs for unskilled persons, and, for that reason, automation is less than would normally be expected.

The coil components—consisting of the tubes; the fin sections, approximately 2 feet long, which have been cut from rolls of aluminum strip and punched to receive the tubes and spacer rings; and aluminum spacer rings, which fit between the tubes and fins— are assembled by hand on a rack and pressed together by a hydraulic ram. After the components are pressed together, expanding mandrels are pulled through the tubes to make a mechanical bond between the coils, fins, and collars. The coils are dipped into an alkaline solution to form an oxide coating on all surfaces for corrosion protection. The water boxes of the coils are made of aluminum and are of welded construction.

Either two or three of the coil sections are joined together to make a "column" and two columns are joined into a "delta", fitted into a supporting steel frame and tested hydrostatically for leaks. The 3-coil delta, approximately 45 feet long, is shipped as an integral unit and handled and erected at the plant site by means of a specially designed carrier.

Dr. Heller and his associates have developed many techniques and devices for control and operation of dry-type towers as a result of over 30 years' experience, and are the holders of over 20 patents applying to dry tower systems.

Dr. Heller has also performed studies of locating a generating plant inside the shell of a natural-draft, dry-type cooling tower in order to take advantage of the uplift from the tower discharge of warm air to disperse stack gases and to overcome inversions, thereby reducing air pollution.

M.A.N. (Maschinenfabrik Augsburg-Nurnberg)

M.A.N. is one of the largest manufacturers of power, heavy mechanical and transportation equipment in West Germany with 7 plants and approximately 50,000 employees. The M.A.N. factory at Nürnberg produces steam and gas turbines, boilers, condensers, heat exchangers and associated steam plant equipment, and has sold a 200-mw, non-reheat steam turbine and dry tower plant ($1,139 \times 10^6$ Btu/hr. exhaust heat) to ESCOM, a large electric utility in South Africa, for their Grootvlei Power Station, which is scheduled for start-up in early 1971.

M.A.N. does not manufacture the cooling coils, but takes responsibility for engineering and procurement of the complete system and the economic selection of the turbine and cooling tower combination. M.A.N. feels that the turbine and dry cooling tower system should, at this stage of the development, be considered as an integral unit rather than selected as two separate components.

M.A.N. will offer either a direct- or indirect-type air cooling system, depending upon the economics of the particular situation studied.

The design of the natural-draft tower as delivered by M.A.N. to ESCOM has the cooling coil tubes in a horizontal position inside the tower and M.A.N. indicates that they expect less wind influence than if the coils were in a vertical position. M.A.N. also states that locating the coils inside the tower shell increases the heat load capability of a tower since the inside area is proportional to the square of the tower diameter, whereas the area available for a circumferential heat exchanger coil installation is only directly proportional to the diameter of the tower.

According to M.A.N., optimization of the dry tower system usually dictates a turbine back pressure above 3.5 inches Hg. They, therefore, eliminate the last row of blades of the standard turbine design and place the shaft bearings outside the low-pressure casings. The turbine capability is maintained over a wide temperature range by providing a second admission point after the initial stages of the turbine and increased boiler capacity for use during periods of high back pressure.

M.A.N. advised that they are prepared to offer steam turbines and dry-type cooling towers up to 1,000 mw in size.

GEA – Gesellschaft Für Luftkondensation

GEA Airexchangers, Inc. of Bochum, West Germany produces finned air-cooling coils for industry and power, and manufactures a direct, air-cooled, condensing system for power stations. GEA is also a licensee for construction of the

Heller system and supplied the dry tower for the 150-mw unit at Ibbenbüren, Germany. GEA has furnished a direct, air-cooled, condensing system for a 160-mw generating unit at Utrillas, Spain (Figure 37), and also furnished the direct, air-cooled, condensing system for the 20.18-mw unit of Black Hills Power and Light Company at Wyodak, Wyoming.

The following estimating figures for approximate cost of the dry-type cooling tower systems in the United States were given to the author by Mr. Hans H. Von Cleve, Chief Engineer of GEA, for the Heller system (indirect) using a natural-draft concrete tower up to 450 feet high (over 450 feet, a steel natural-draft tower would be used). With a distance of 300 feet between the tower and the turbine, the cost is estimated to be:

$$\$520{,}000 \times \frac{\text{heat load, } 10^6 \text{ Btu/hr.}}{(\text{ITD, } ^\circ\text{F})^{1.25}}$$

The above cost is for an erected system covering all condensing system components from the turbine flange outward, tower, pumps, piping, foundations, etc. According to Mr. Von Cleve, GEA is prepared to offer a cooling tower system up to 1,000 mw in size, and has actually quoted 450- and 900-mw sizes to United States utilities on the above basis.

The independent cost estimates made for the 800-mw, fossil-fueled plant used in the computer program of this report corresponded closely with the foregoing GEA cost estimating formula.

Mr. Von Cleve's estimate for a direct, air-cooled, condensing system with mechanical draft for sizes up to 200 or 300 mw is:

$$\$210{,}000 \times \frac{\text{heat load, } 10^6 \text{ Btu/hr.}}{\text{ITD, } ^\circ\text{F}} \quad \text{— Basic Estimate}$$

to which must be added:

0.12 x the basic estimate for the steel structure
0.08 x the basic estimate for the exhaust trunk
0.12 x the basic estimate for erection

The required land area for the direct system is:

$$200 \times \frac{\text{heat load, } 10^6 \text{ Btu/hr.}}{\text{ITD, } ^\circ\text{F}} \quad \text{sq.ft.}$$

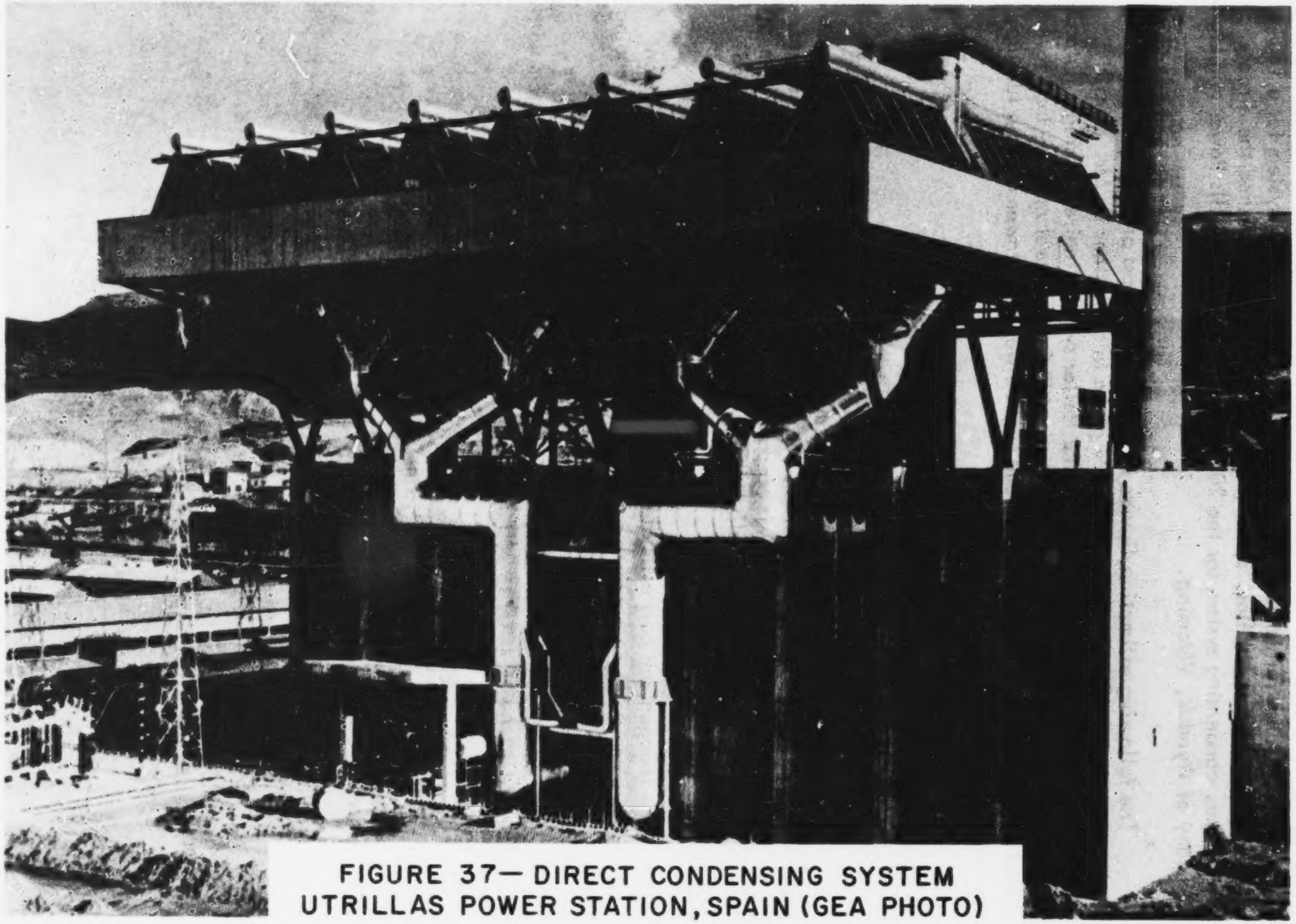


FIGURE 37— DIRECT CONDENSING SYSTEM
UTRILLAS POWER STATION, SPAIN (GEA PHOTO)

Installed fan power is:

$$250 \times \frac{\text{heat load, } 10^6 \text{ Btu/hr.}}{\text{ITD, } ^{\circ}\text{F}} \quad \text{kw}$$

The power consumption of the fans is approximately 2 percent of the gross output. Mr. Von Cleve estimates that the cost difference between a plant equipped with a conventional tower and a GEA direct, air-cooled, condensing system is approximately $5 to $6 per kw for sizes up to 300 mw.

For sizes larger than approximately 200 to 300 mw maximum, GEA would offer the indirect system only.

English Electric Company

English Electric Company, now English Electric – AEI Turbine Generators Limited as a result of a recent merger, has undertaken development work on the indirect dry tower system for approximately 10 years. English Electric has a license from Transelektro for marketing and constructing the Heller tower in the United Kingdom and furnished the dry tower for the 120-mw unit at Rugeley Station, which went into service in 1961. At that time, it was believed that most of the large generating stations in England would be constructed at mine sites and at inland locations close to fuel supplies, and it was anticipated that cooling water make-up for evaporative towers would soon be a problem. However, it now appears more likely that future large generating plants in England will be nuclear or oil-fired and located on the sea coast, with the result that cooling water for once-through condensing systems will not be a problem and the need for dry cooling towers in England may not be as imminent as had once been thought.

Conferences were held with Mr. W. H. P. Wolff, Technical Director of the Willans Works at Rugby, now Director of British Nuclear Design and Construction Ltd., and Messrs. D. W. Crane, P. J. Christopher and J. L. Daltry, who have been engaged in the dry cooling program.

Studies made by English Electric indicate that capital costs are increased approximately $12 to $14 per kw for a dry tower plant, and that the average bus-bar costs, taking into account fixed costs and fuel costs, are increased approximately 6 percent as compared to an evaporative tower system. English Electric considers that the components of a dry tower system cost about 1-1/2 times the cost of an equivalent wet tower system.

English Electric have concluded that a fully cost-optimized dry cooling tower scheme would require a somewhat higher back pressure on the turbine than with a water-cooled condenser. In the majority of cases that they have studied, the optimized back pressure increases relative to wet tower cooling by an amount broadly in the region of 1.0 inches Hg. For such a case, they would consider that modification of the last-stage blades of a turbine designed for other cooling systems would be practicable.

In the case of Rugeley, the turbine exhaust pressure for the dry cooling tower was specified to be the same as for the other turbines in the station associated with evaporative towers. In this respect, Rugeley is not a cost-optimized scheme and the question of turbine design for higher back pressure did not arise.

Brown Boveri Corporation

A visit was made to the Brown Boveri plant at Baden, Switzerland to discuss the operation of large turbine generators at back pressures higher than 3.5 inches Hg. Brown Boveri is one of the major manufacturers of power-generating equipment in the industry and is presently constructing turbine-generator units up to 1,300 mw. Discussions were held with Mr. W. Hossli, Head of the Turbine Department Design & Calculation; Mr. H. Mühlhauser, Head of the Turbine Performance Section; and other Brown Boveri engineers.

Brown Boveri is interested in the use of dry cooling towers for generating plants and has participated in studies of dry towers for utilities. Brown Boveri does not produce dry tower equipment, but has used Dr. Heller as a consultant.

Brown Boveri believes that if there is a demand for a large number of turbines to operate with dry cooling towers at high back pressures, a new design will be developed. They estimate that a turbine designed for 6 inches Hg back pressure would cost approximately 15 percent less than a turbine designed for 2 inches Hg back pressure. However, a high-back-pressure turbine would probably not be designed until there were enough units foreseen to absorb the development costs. Until that time, a standard modified design having a shorter last-stage blade (eliminating the last stage) or reducing the number of low-pressure turbines could be used.

United States Turbine Manufacturers

Discussions and correspondence were held with the two major manufacturers of large turbine generators in the United States—the General Electric Company and the Westinghouse Electric Corporation—to obtain their respective opinions as to the feasibility of operating large turbines at high back pressures. The results of these discussions are covered in Section III of this report.

General Electric believes that until there is a sufficient demand for a specially designed high-back-pressure turbine, modifications would be made to existing units for operation at back pressures up to 15 inches Hg.

Westinghouse advises that it is possible to design and manufacture a turbine for high back pressure application. Modifications of an existing or standard design unit may or may not be economically practical, depending upon the particular unit involved.

Hudson Products Corporation

Hudson Products Corporation of Houston, Texas has manufactured air-cooled heat exchanger equipment for chemical and refinery industries for over 25 years. Today, they and their foreign licensees are the world's largest manufacturers of industrial air-cooled heat exchangers. At one time, they also manufactured conventional mechanical-draft, wet-type cooling towers for the process and power industries. As a result of extensive research and development in air-cooled heat transfer surfaces, hyperbolic tower shells and large fans designed specifically for power plant application, Hudson Products Corporation is now offering a dry-type cooling tower to the utility industry. Hudson Engineering Corporation, a subsidiary of J. Ray McDermott, as is also Hudson Products Corporation, is prepared to offer a dry cooling tower system package completely engineered and installed. The system starts at the turbine exhaust flange and includes the direct-contact condenser, circulating water pumps, dry-type cooling tower, piping, valves and controls.

Conferences were held with Mr. Ennis C. Smith, Vice President and General Manager, and Mr. Michael W. Larinoff, Vice President, to obtain cost estimating information and tower performance data for use in this report. An excellent summary of Hudson's performance and economic studies is contained in "Power Plant Siting, Performance and Economics with Dry Cooling Tower Systems" by Smith and Larinoff, presented at the 1970 American Power Conference (13).

The Marley Company

The Marley Company of Kansas City, Missouri is one of the largest manufacturers of evaporative-type cooling towers in the world, with operations in many foreign countries as well as in the United States.

Conferences were held with Marley engineers, including Mr. Joe Ben Dickey, Jr., Vice President of Engineering; Mr. J. O. Kadel, Vice President of Major Projects; Mr. Robert E. Cates, Senior Evaluations Engineer; Mr. John A. Nelson, Senior Metallurgist; Mr. Edward P. Hansen, Vice President of Marfab Radiator Division; and Mr. Joel Blake, Consulting Engineer to DriTower Committee. During these conferences, much information was obtained for use in this report. The following excerpt, taken from "Managing Waste Heat with the Water Cooling Tower", by Joe Ben Dickey, Jr. and Robert E. Cates of the Marley Company, sums up Marley's work on the dry-type cooling tower:

"DRITOWERS*

"For the past four years The Marley Company has contributed its Division Managers of Major Projects; the L. T. Mart Company of England; Marfab, its Radiator Company; Engineering Division; its Senior Evaluations Engineer; its Senior Metallurgist; and a distinguished alumnus of an experienced operating company as a team constituting its DriTower Committee. The first year, five exterior corrosion induced draft coil sample demonstrations were installed at operating plants of the American Electric Power and General Public Utilities Systems. The second year internal corrosion test tables were installed with all manner of suitable tube alloys circulating actual plant deionized water. In the second winter extensive outside freeze tests on a full scale model were conducted at the Marley Research Laboratories. During the third year the disappointing results of the first exterior corrosion studies resulted in the commissioning of a second generation induced draft wrapped tube study. Throughout this three year period continuous re-evaluation of all American and foreign wrapped fins and core sections were conducted in the dry lab heat transfer wind tunnel in Kansas City. Foul factors, tube spacing, and boundary-layer turbulence were analyzed. While outside of the scope of this paper, the authors may briefly comment that for the freezing North American latitudes the natural draft hyperbolic Dry Tower of the style built abroad would have freezing problems, start-up and shut-down problems, and corrosion problems that would magnify both the operating cost and technique beyond the conception of present market acceptability in America. In the southern climates of the United States, to permit the difference between dry bulb and steam operating temperature, the economic usage of a DriTower on large power plants would require turbines and open condensers not foreseeably available in this decade. The vexing problem of managing the large steam quantities in the generating plant sizes planned in future years, caused this Committee to abandon the possibility of direct steam condensers of any consequence in North American latitudes. Induced Draft DriTowers scaled-up in size from those designs commonly proven most operable and most economic in the hydrocarbon and petrochemical industries of America were found by the DriTower Committee to offer the best interim solution in this country. Even then, the designer must be prepared for considerable study and attention to controls, signal monitors, dampers, and dumping mechanisms which would result in equipment having much higher risk of problems during a rapid scram, and much more manpower devoted to fine tuning than an American market will readily digest. As a final project in calendar 1969, the Committee was a partner of a prominent Eastern

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* DriTower is Registered Trademark.

Architect-Engineer in the development of a full-fledged proforma plant design using both Natural Draft and Induced Draft DriTower on an 800 megawatt machine. The Committee would observe that these estimates are extremely complicated because of many parenthetical matters that are completely foreign to a normal estimate. Because of the vast original work required for a rigorous study, simple rule of thumb estimates on Induced Draft units, only, are recommended. The committee's observations, together with information now available on pioneer installations in the Eastern hemisphere teach us that the materials section, the maintenance labor, and the extensive controls required by a Northern DriTower plant, form a very sober undertaking. For very clean nonfreezing sites, given very low cost fuel, economic transmission, and tax plus ecological incentive, DriTowers will deserve study as rotating and condensing hardware becomes suitable."

Ingersoll-Rand Company

Ingersoll-Rand Company of Phillipsburg, New Jersey is making studies of the cost of direct-contact condensers for use with dry-type cooling towers. Cost estimating data of direct-contact condensers furnished by Ingersoll-Rand were used in the optimization studies in this report.

GKN Birwelco Limited

GKN Birwelco Limited is a subsidiary of Guest, Keen and Nettlefolds, a large English-based international engineering group. They are specialists in the design and execution of substantial contracts involving heat transfer equipment and were responsible for the complete process, mechanical and civil design, purchasing, inspection and construction of the 200-megawatt dry cooling tower which is now being commissioned for Escom, a large electricity utility in South Africa. They offer complete construction of both direct and indirect condensing systems using natural or mechanical draft systems. GKN Birwelco, through its New York subsidiary, GKN International, Inc., offers these installations using complete supply of materials and services from United States manufacturers and is currently performing studies using United States subcontractors for dry cooling towers up to 1,000 megawatts in size.

GKN Birwelco uses cooling sections in a horizontal position inside the base of the natural draft tower shell rather than the vertical arrangement previously described. They have performed wind tunnel tests at the National Physical Laboratory which they state have shown enhanced performance of the horizontal sections during windy conditions.

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SECTION IX

OPTIMIZATION PROGRAMS

Introduction

The state of the art of dry cooling for steam-electric generation is such that relatively little information is presently available to assist electric utilities in evaluating the economics of dry cooling. In contrast, the various methods of wet cooling have been in use by the electric utility industry long enough so that the sizing, performance and cost of the economically optimum, or near optimum, cooling system can be established with relative ease.

A method of analysis and a computer program were developed to determine the cost and performance of a range of sizes of dry-type cooling systems for specific sets of conditions and to select the economically optimum size for those conditions. The measure of dry-type cooling system size used in the analysis is the initial temperature difference (ITD) which is defined as the temperature difference between the turbine exhaust steam and the ambient air. The concept of initial temperature difference is discussed in Section II of this report. The specific conditions which affect the selection of the economically optimum dry cooling system include such factors as the relationships of performance and capital cost to ITD, the fixed-charge rate, fuel cost, air temperature, the amount of generating capability lost at high ambient air temperature, and the cost of replacing the lost capability. These various factors and their effects on the optimization of the dry cooling system are discussed in detail later in this section.

Two computer programs were developed to facilitate the analysis. The first program determines the optimum tower size for a given ITD. The second chooses the optimum ITD and, consequently, tower size with consideration given to all costs of construction and operation.

The method of analysis and the computer programs yield information as to the size, cost and performance of the economically optimum dry cooling system for a specific set of conditions, but does not provide the information necessary to compare the relative economics of dry cooling versus other cooling methods. Although it was not within the scope of this study to compare the economics of dry cooling with other methods, some preliminary economic comparisons of dry cooling systems versus wet tower cooling systems were made in order to indicate the factors which must be considered in such a comparison and to establish the order of magnitude of the cost differences which may result if a dry-type cooling system is utilized in lieu of a conventional wet tower cooling system. These preliminary economic comparisons are discussed in Section XII of this report.

Method of Analysis and Description of Tower Optimization Program

An important preliminary step in determining the economically optimum ITD for a dry-type cooling system is the selection of the design parameters of the cooling tower to choose the optimum design to use in the studies. The basis for the method of optimizing the tower design was described in Section III — Performance. The purpose of this optimization is to determine the most economically optimum balance of tower and equipment to achieve the lowest construction capital cost.

Since a given ITD can be achieved by a number of combinations of air flow, water flow, water temperature range and approach to the ambient air temperature, it was necessary to evaluate their interaction. Five parameters are considered in the tower optimization. They are range, amount of heat rejected, quantity of water flowing in the coils, ambient air temperature and tower elevation. Output of the program is tower height, stack diameter at the top of the tower, diameter at its base and its cost.

For a given heat rejection at a given ITD, the range and water flow are varied to provide the minimum cost of the tower.

The cost estimate is divided into tower structure, condenser, piping and controls. The cooling coils are included with the tower cost. A sample computer printout is shown in Table 7.

The basic capital cost of the dry cooling systems which were developed are assumed to be average United States costs. Applicable construction cost indexes have been analyzed and capital cost multipliers have been determined for each of the 27 sites to approximately reflect changes in capital costs which may be expected from area to area.

In addition, structural analyses of the natural-draft cooling tower indicate that the capital cost of the natural-draft, dry-type cooling system should be increased by about 2 percent to reflect the higher cost tower structure necessary in areas subject to hurricanes. This 2 percent adjustment is reflected in the capital cost multipliers applied to natural-draft systems for two of the sites investigated.

A procedure, comparable to the one described above, was followed in the optimization of mechanical-draft towers.

The physical sizes of the dry cooling towers corresponding to the ITD values are shown in Figure 38 for natural-draft towers and Figure 39 for mechanical-draft towers.

TABLE 7
COMPUTER PRINTOUT – NATURAL-DRAFT COOLING TOWER SYSTEM–SIZING AND COSTING PROGRAM

```
DRY COOLING TOWER SIZING AND COST EVALUATION

DESIGN PARAMETERS

            ITD =     60       RANGE =    30
            HEAT REJECTION  =    4.0E+09
            WATER FLOW PER HOUR  =    2.2E+05
            AMBIENT AIR TEMP  =    50       ELEVATION  =    3000

TOWER SIZING

            TOWER HEIGHT  =  539.1
            UPPER DIAMETER  =  346.8
            BOTTOM DIAMETER  =  450.6
            GALLONS PER MINUTE  =  266549

COST EVALUATION
    TOWER STRUCTURE

            STACK COST                    2120830
            SHED COST                      493872
            COIL COST                     4408000

            TOTAL STRUCTURE                                7022702

    CONDENSER

            CONDENSER COST                                  832000

    PIPING, VALVES, ETC.

            PIPE COST                     1306835
            VALVE COST                     833500
            PUMP COST                     1200000
            FILLER PUMP COST                40000
            STORAGE TANK COST               28720

            TOTAL PIPING FACILITIES                        3409054

    CONTROLS

            CONTROL COST                                    500000

COMPLETE TOWER FACILITIES

            TOTAL TOWER COST                              11763756

TOTAL TOWER COST AND CONTINGENCIES                        14704696
```

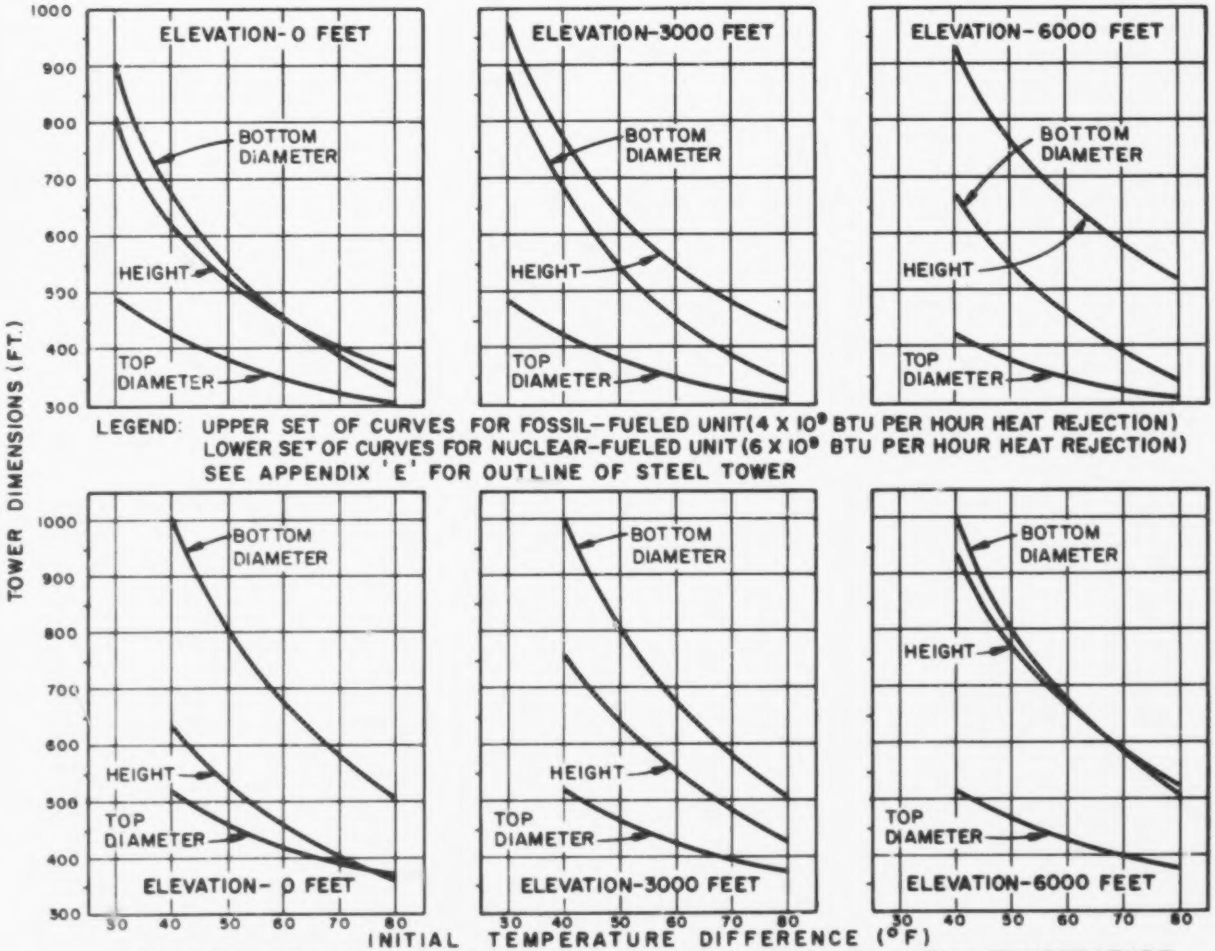


FIGURE 38—COOLING TOWER DIMENSIONS AS A FUNCTION OF INITIAL TEMPERATURE DIFFERENCE AND ELEVATION FOR NATURAL-DRAFT COOLING TOWERS—STEEL AND ALUMINUM CONSTRUCTION—800 MW GENERATING CAPACITY

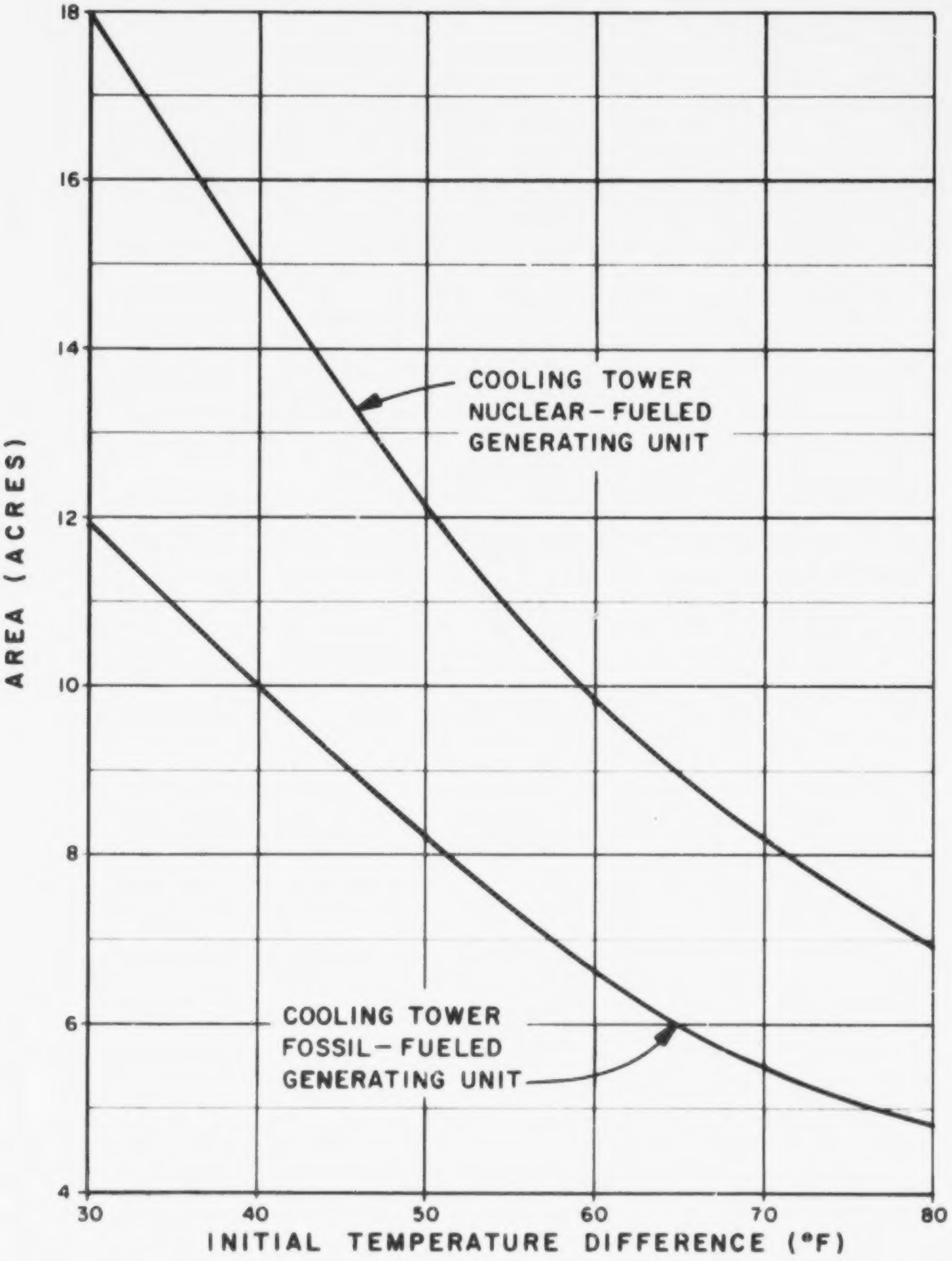


FIGURE 39 — GROUND AREA REQUIREMENT AS A FUNCTION OF INITIAL TEMPERATURE DIFFERENCE FOR MECHANICAL-DRAFT, DRY COOLING TOWERS- 800 MW UNIT

Figure 40 shows the capital cost, in dollars per kw, of natural-draft and mechanical-draft systems for both fossil- and nuclear-fueled plants. The costs are shown for ITD values of 30° to 80°F and for ground elevations of 0, 3,000 and 6,000 feet.

Factors Affecting the Economic Optimization of Dry-Type Cooling Towers

The economically optimum dry cooling system for a specific set of conditions is that which results in the lowest annual cost. The annual cost must reflect all costs incurred on an annual basis, such as operation, maintenance, total plant fuel costs and the annual capital costs.

The performance of dry cooling systems has been discussed in Section III of this report and the key factors affecting the economic optimization stem from those performance characteristics and the capital cost of the cooling system. These key factors are:

1. The effect of increasing the ITD and/or the ambient air temperature is to increase the temperature of the turbine exhaust steam and, therefore, the turbine back pressure. An increase in turbine back pressure results in poorer fuel economy and in loss of generating capability.

2. The physical size and, therefore, the capital cost of the dry-type cooling system decreases with increasing ITD.

The combination of the above factors indicates that a dry cooling system could be: 1) a low-ITD, high-capital-cost system with good fuel economy and little or no loss of generating capability at high ambient air temperatures; or, 2) a high-ITD, low-capital-cost system with poorer fuel economy and a significant loss of generating capability at high ambient air temperatures; or, 3) some intermediate-size cooling system. The economically optimum dry cooling system for a specific location and specific set of conditions must reflect the effect of a number of variables. Those variables which affect the economic optimization are discussed below.

Performance related to ITD. The effect of increasing the ITD is to increase the exhaust steam temperature for a given air temperature and, therefore, the turbine back pressure. This results in poorer fuel economy and loss of generating capability.

Capital cost of the dry cooling system. The physical size and the capital cost of the dry cooling system decrease with increasing ITD.

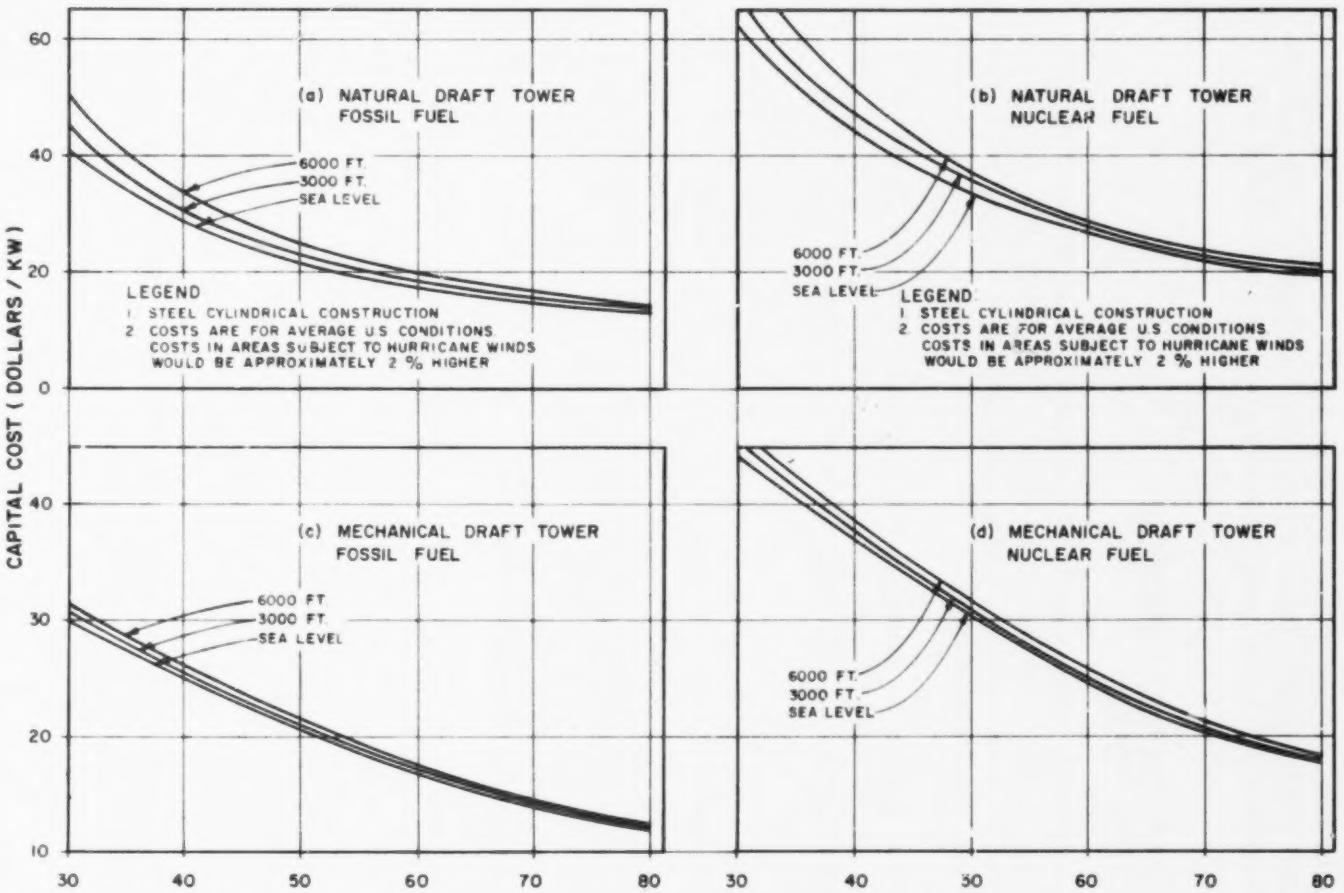


FIGURE 40—RELATIONSHIP OF DRY COOLING SYSTEM CAPITAL COST TO ITD AND ELEVATION—800 MW GENERATING PLANT (1970 COST LEVEL)

Elevation. The effect of increasing ground-level elevation is to increase the capital cost of the dry cooling system since the reduced air density makes it necessary to move a greater volume of air past the cooling elements in order to achieve the same mass flow rate of air.

Fixed-charge rate. The fixed-charge rate is a percentage rate applied to the capital cost which reflects the following items as defined by the Bureau of Power of the Federal Power Commission (40):

1. Interest, or cost of money.
2. Depreciation, or amortization.
3. Interim replacements.
4. Insurance, or payments in lieu of insurance.
5. Taxes (federal, state and local), or payments in lieu of taxes.

The effect on the economic optimization of an increase in the fixed-charge rate is to give more weight to capital costs and less weight to annual operation, maintenance and fuel costs.

Ambient air temperatures. The effect of higher ambient air temperatures is to increase the turbine back pressure resulting in poorer fuel economy and loss of generating capability. The full range of annual air temperatures at the site affect the fuel economy, but it is the extreme high temperature which has the more significant economic effect. The extreme high temperature, in combination with the cooling system ITD and the turbine characteristics, sets the maximum loss of generating capability which would be experienced during the year.

Fuel costs. The effect of increasing the unit cost of fuel is to increase the weight given to fuel economy and decrease the weight given to capital cost considerations. Therefore, increasing the fuel cost would tend to reduce the optimum ITD, or, in other words, would tend toward a higher capital cost cooling system.

Turbine performance. The shape of the turbine performance curve of heat rate versus back pressure is important in that it affects the relative importance of fuel economy and loss of generating capability. As shown on Figure 28 in Section III of this report, a conventional turbine modified to operate at high back pressures would have a relatively low heat rate at low back pressures, and would have a poorer heat rate with resulting loss of both economy and generating capability at high back pressures. On the other hand, the high-back-pressure turbine would have a poorer heat rate at low back pressures than the modified conventional turbine, but the loss of capability at high back pressures would not be as pronounced.

Auxiliary power requirements. The power requirements for pumps and fans decrease with increasing ITD.

Replacement of capacity losses. Since some loss of generating capability can be expected at high ambient air temperatures with a dry cooling system, the replacement of this capability is an important consideration in the economic analysis. The relative importance of this capability loss may vary from area to area and would, of course, be of the greatest concern in an area where the annual peak electrical demand occurs in the summer rather than in the winter. In this instance, the capability lost would need to be replaced from some other source. A utility having a winter peak system demand would not be as much affected by the loss of generating capability on a hot summer day, with respect to meeting its own demands, but may still be economically interested in the lost capacity since that utility may have the opportunity to sell surplus capacity to other interconnected systems. Once it has been determined whether or not the replacement of the lost capacity is necessary, then the cost of replacing that capacity must be determined. In the optimization, the economic impact of the capacity loss increases as the cost of replacing that capacity increases. Therefore, the significance of lost capacity is much greater if the lost capacity is replaced at a capital cost of $150 per kw than if it is replaced at a capital cost of $100 per kw.

Method of Analysis and Description of the Economic Optimization Program

The method of analysis which was applied in the determination of the economically optimum dry cooling system for various conditions is based on an analysis of all costs which would be affected by the choice of size, or ITD, of the dry cooling system. Therefore, the costs reflected are the plant fuel cost and all costs related to the dry cooling system which is defined as those facilities from the turbine flange outward. These facilities would include the condenser, the cooling system piping, water storage facilities, pumps, valves, controls, recovery turbine if used, and the cooling tower with its heat exchanger equipment. The analysis does not include consideration of the other generating plant costs since those costs would not vary with the selection of the dry cooling system ITD. Also included in the analysis is the economic consideration of the generating capability lost at high ambient air temperatures.

For the purposes of this analysis, fossil- and nuclear-fueled generating plants of 800-mw size were assumed. The results of the analysis, as evaluated on a cost-per-kw basis, should be generally applicable to generating plants in the size range of 600 mw to 1,000 mw, or perhaps over a somewhat larger range of sizes.

It is recognized that it may not be practical to build an 800-mw unit at some of the sites which were selected for analysis. The sites included in the analyses were not selected as being particularly likely sites for the construction of an 800-mw unit, but were selected to represent a variety of air temperature, elevation and fuel cost conditions so that the effect of these factors on the economic optimization could be analyzed.

The method of analysis and the computer program were developed to handle both nuclear- and fossil-fueled generating units and both natural-draft and mechanical-draft dry cooling systems. Although a dry-type cooling tower has not yet been used with a nuclear plant, there is a great need for such combination due to the relatively large amount of waste heat rejected by the turbine and consequent heat addition to natural bodies of water as compared to fossil-fueled plants. However, before a dry-type tower can be built with a nuclear plant, important questions involving shielding requirements, necessitated as a result of the direct mixing of turbine exhaust steam and circulating water, must be resolved by the agencies having jurisdiction over these matters.

On the basis of information obtained as to the sizing and performance characteristics of existing dry cooling systems, it was determined that the analysis should cover a range of ITD values and that range was established as 30^{o}F to 80^{o}F. The design value of ITD for a dry cooling system is related to a specific value of heat rejection, as discussed in Section II of this report. For this analysis, the design ITD is that which occurs at a nominal heat rejection of 4×10^9 Btu per hour for a fossil-fueled plant, and at a nominal value of 6×10^9 Btu per hour for a nuclear-fueled plant. As shown in Figure 24 of Section III of this report, the heat rejection capability of the cooling system varies with turbine back pressure. The fossil plant nominal heat rejection value of 4×10^9 Btu per hour and the nuclear plant nominal heat value rejection of 6×10^9 Btu per hour both occur at a turbine back pressure of approximately 8 inches Hg. The following tabulation shows the rates of heat rejection requirements for the fossil and nuclear plants for several specific back pressures. The heat rejection values shown in the table are based on full throttle flow performance and indicate the reduced generating capability at elevated back pressures.

TABLE 8

Heat Rejection Versus Back Pressure for an 800-Mw Generating Unit (Full Throttle Flow Performance)

Turbine Back Pressure (in. Hg)	Fossil		Nuclear	
	Output (mw)	Heat Rejection (10^9 Btu/hr.)	Output (mw)	Heat Rejection (10^9 Btu/hr.)
2.0	809	3.80	814	5.68
4.0	796	3.85	790	5.76
6.0	771	3.93	751	5.89
8.0	750	4.01	718	6.01
10.0	728	4.08	692	6.09
12.0	709	4.14	672	6.16
14.0	692	4.20	655	6.22

The performance of the cooling system and the turbine have been discussed previously in Section III of this report. Figure 24 of Section III illustrates the interrelationship of the tower and turbine performance curves.

As a result of preliminary analysis, a standard design assumption which results in a dry-type cooling system very close in cost to the cooling system that would be selected by much more detailed analysis was established. The more detailed analysis would consist of an evaluation for each set of conditions of the economic effect of varying range, approach, air flow and water flow. The approach of this study has been to establish the over-all economics of dry cooling systems for a large number of combinations of conditions. This has been accomplished. Once a specific site has been selected for a detailed analysis, it would, of course, be necessary to thoroughly investigate the effect of these other variables in order to refine the cooling system design.

The economic optimization program consists of an analysis of the annual costs which are affected by the size, or ITD, of the dry cooling system for each 1°F differential of ITD between ITD values of 30°F and 80°F. The costs evaluated are the annual capital cost of the dry cooling system, calculated as the capital cost times the fixed-charge rate; the annual operation and maintenance cost of the dry cooling system; the annual fuel cost of the 800-mw unit; the annual cost of power and energy required by the cooling system pumps and fans; and the annual cost of replacing the capacity and energy lost due to high-back-pressure operation. These annual costs are summed and the minimum value of that sum within the range of ITD values analyzed defines the economically optimum size of dry cooling system.

Each of the analyses made reflects certain specific assumptions as to site elevation, annual air temperatures, season of system peak demand, type of fuel (fossil or nuclear), fuel cost, turbine characteristics, method of operation of the generating plant, type of cooling tower (natural draft or mechanical draft), cooling system capital costs, fixed-charge rate, cost of auxiliary power and energy, and cost of replacing lost capacity.

For the purposes of the economic optimization analyses which are summarized in Section X of this report, 27 sites within the United States were chosen to represent a range of air temperature conditions, ground-level elevations and fuel costs. The annual air temperature data were obtained from the U.S. Weather Bureau Bulletin 82, "Climatography of the United States" (41), which summarizes the frequency of occurrence of air temperatures. The ground-level elevations used in the analyses were the weather station elevations rounded to the nearest 100 feet above sea level.

Analyses were made for each of these 27 sites for both fossil- and nuclear-fueled generating units, natural-draft and mechanical-draft dry cooling systems, with fixed-charge rates ranging from 8 percent to 18 percent and a range of fuel costs.

It is believed that the range of fixed-charge rates of 8 percent to 18 percent represents the range of values which would be applicable to electric utilities in the United States for new construction.

The fuel costs selected are generally representative of existing fuel costs. In some cases, the highest value of fuel cost investigated may be somewhat higher than current fuel costs, but no attempt has been made to predict future fuel prices, just as no attempt has been made to predict the future capital cost of the dry-type cooling system. In general, all costs used in the analyses are current (1970) prices.

In all cases, an allowance for the operation and maintenance cost of the dry-type cooling system was estimated at 1 percent of the capital cost of the dry-type cooling system.

For this analysis, it was assumed that the generating plant would operate 7,500 hours per year, 50 percent of that time at full throttle and 50 percent of that time at 75 percent load, which is equivalent to 600 mw. The annual generation required by the system from this 800-mw unit under the assumptions stated would, therefore, be 5,250,000 mwh. In the analyses, the total energy production of the 800-mw unit was computed, reflecting both capacity gains and losses. The energy gains and losses were then computed and considered in the economic analysis. A credit for the energy gains, reflecting the fuel cost of generating that energy, was calculated and the cost of replacing the energy losses was also calculated, as described later. The net result is that the analyses reflect the cooling system costs and the costs of supplemental peaking generation necessary to produce a combined output of 800 mw at the high air temperatures, as defined later, and a combined energy output of 5,250,000 mwh per year.

As discussed previously, the economic impact of the loss of generating capability at high ambient air temperatures is more significant if the period of maximum demand occurs in the summer rather than in the winter. For the basic analyses, a summer peak was assumed for all sites except the one located in Alaska. Some supplemental analyses were made to indicate the effect of assuming a winter peak. It is recognized that not all of the sites investigated lie in summer peak areas, but it was assumed that utilities constructing plants in these areas would have the opportunity to market excess generating capability to other utilities in the summer and, therefore, would have an economic interest in the loss of generating capability.

For the purposes of these analyses, the loss of generating capability was evaluated at the ambient air temperature for the site which is equalled or exceeded, on the average, only 10 hours each year. It would not be reasonable to evaluate the lost capacity for the extreme maximum temperature. The peak electrical system load may not occur at the coincidental time of maximum hourly temperature. In fact, the use of the temperature equalled or exceeded only 10 hours each year may be somewhat severe and it may be reasonable to evaluate that loss of capability at a lower air temperature. A possible alternative to the 10-hour temperature would be that temperature which is equalled or exceeded only 1 percent of the time during the 4-month period June through September. That temperature duration would be 1 percent of 2,928 hours, or about 29 hours. This temperature duration is commonly used in the analysis of wet-bulb temperatures for wet cooling tower design.

It was assumed that the loss of capability which was experienced at the air temperature which is equalled or exceeded only 10 hours per year would be replaced by generation from another source. The basic analyses are based on the assumption that the source of the replacement capacity would be peaking units having a capital cost of $100 per kw. The loss of energy generated due to high-back-pressure conditions would also be replaced by these peaking units. The cost of energy from these peaking units reflects a heat rate of 15,000 Btu per kwh and, in most cases, a fuel cost of $0.40 per million Btu. In some cases, it was assumed that natural gas would be available to operate the peaking and, therefore, a somewhat lower fuel cost was assumed.

The cost of the auxiliary power and energy required for the cooling system pumps and fans was calculated assuming that incremental steam plant capacity could be provided for a cost of $150 per kw for fossil-fueled units and $225 per kw for nuclear-fueled units, and that the energy cost for the auxiliaries would be the average fuel cost in mills per kwh of the 800-mw plant plus an allowance for operation and maintenance costs of 0.1 mills per kwh.

Perhaps the method of analysis can best be described by summarizing the results of one of the analyses as presented on the computer printout. Table 9 shows the computer printout for a natural-draft dry cooling system associated with a fossil-fueled plant at Burlington, Vermont for a plant fuel cost of 25¢ per million Btu, a peaking fuel cost of 40¢ per million Btu and an annual fixed-charge rage of 15 percent.

Referring to Table 9, the first column shows the initial temperature difference (ITD). The second column shows the gross energy generation of the 800-mw unit reflecting both the capacity gains at back pressures less than 3.5 inches Hg and capacity losses at back pressures above 3.5 inches Hg. The third column shows the amount of the excess energy due to operation at back pressures under 3.5 inches Hg. The fourth column shows the energy associated with capacity losses at back pressures above 3.5 inches Hg. The column headed "Auxiliary Energy" shows the annual energy requirement of the cooling system pumps. The column headed "Loss of Capacity" shows the capacity lost at the air temperature which is equalled or exceeded only 10 hours each year. The column headed "Maximum Auxiliary Power" shows the maximum capacity required for the cooling system pumps. For the mechanical-draft analyses, the auxiliary power and energy requirements would, of course, also reflect the cooling system fan requirements.

The next seven columns of Table 9 show annual cooling system costs. The column "Annual Capital and O&M Cost of Dry Cooling System" is the capital cost of the dry cooling system multiplied by the fixed-charge rate plus the 1 percent allowance for operation and maintenance cost. In this case, the column is computed at the capital cost times 16 percent. The next column shows the total annual fuel cost of the 800-mw unit and reflects the gross energy generation as shown in column two. The column headed "Credit for Excess Energy" is a fuel-cost credit related to the excess energy amounts shown in the third column. This credit reflects the fuel cost of energy generated by the 800-mw unit. The column headed "Capacity Penalty Cost" reflects the costs of replacing both the capacity and energy losses due to operation and at back pressures above 3.5 inches Hg. The column headed "Auxiliary Cost" reflects the cost of providing the power and energy necessary to supply the cooling system pumps.

The total annual cost in dollars is the sum of the preceding five columns and the total annual cost in mills per kwh is that sum divided by 5,250,000 mwh. The optimum ITD is that which produces the lowest total annual cost and in this case is 57°F, which results in a total annual cost of $15,203,529, equivalent to 2.8959 mills per kwh.

TABLE 9— COMPUTER PRINT-OUT, ECONOMIC OPTIMIZATION,800 MW,FOSSIL-FUELED GENERATING UNIT, NATURAL-DRAFT TOWER, BURLINGTON , VERMONT

CAPITAL COST FACTORS: PLANT - 15 0/0 PEAKING CAPACITY - 15 0/0 AUXILIARIES - 15 0/0
PLANT FUEL COST - 25 CENTS/10**6 BTU PEAKING FUEL COST - 40 CENTS/10**6 BTU
PEAKING CAPITAL COST - 100 $/KW AUXILIARY CAPITAL COST - 150 $/KW

INIT. TEMP. DIFF. (DEG) (F)	GROSS ENERGY 800 MW UNIT (MWH)	EXCESS ENERGY DUE TO EXTRA CAPACITY (MWH)	CAPACITY PENALTY ENERGY (MWH)	AUXILIARY ENERGY (MWH)	LOSS OF CAPACITY (KW)	MAXIMUM AUXILIARY POWER (KW)	ANNUAL CAPITAL AND O+M COST OF DRY COOLING SYSTEM ($)	ANNUAL FUEL COST OF 800 MW UNIT ($)	CREDIT FOR EXCESS ENERGY ($)	CAPACITY PENALTY COST ($)	AUXILIARY COST ($)	TOTAL ANNUAL COST OF COOLING SYSTEM AND TOTAL PLANT FUEL ================== ($)	(MILL/KWH)
30	5285897	35901	3	93375	249	12450	5004988	11969738	80808	4055	500907	17398879	3.3141
31	5285716	35723	7	90624	1077	12083	4825202	11969991	80411	17490	486156	17218428	3.2797
32	5285511	35521	11	87966	1942	11729	4651822	11970270	79961	31525	471919	17045574	3.2468
33	5285289	35303	15	85401	2846	11387	4484847	11970573	79472	46193	458171	16880312	3.2153
34	5285049	35067	19	82929	3791	11057	4324277	11970904	78944	61527	444914	16722677	3.1853
35	5284789	34818	29	80550	4819	10740	4170112	11971255	78385	78244	432169	16573395	3.1568
36	5284495	34548	53	78264	5944	10435	4022353	11971636	77781	96610	419915	16432735	3.1300
37	5284161	34239	78	76071	7144	10143	3880999	11972053	77089	116199	408175	16300338	3.1048
38	5283799	33903	104	73971	8426	9863	3746051	11972508	76335	137124	396925	16176272	3.0812
39	5283413	33544	131	71964	9791	9595	3617507	11973001	75529	159400	386165	16060545	3.0592
40	5282994	33166	171	70050	11242	9340	3495369	11973532	74680	183149	375918	15953289	3.0387
41	5282538	32808	271	68229	12786	9097	3379636	11974096	73878	208757	366162	15854774	3.0200
42	5282023	32398	375	66501	14429	8867	3270309	11974704	72959	236000	356920	15764975	3.0029
43	5281468	31953	485	64866	16160	8649	3167387	11975365	71958	264701	348168	15683663	2.9874
44	5280876	31476	600	63324	17939	8443	3070870	11976081	70886	294213	339907	15610184	2.9734
45	5280236	30959	724	61875	19797	8250	2980758	11976852	69724	325053	332160	15545098	2.9610
46	5279550	30536	986	60519	21737	8069	2897052	11977681	68773	358052	324903	15488916	2.9503
47	5278787	30049	1262	59256	23690	7901	2819750	11978568	67681	391350	318161	15440149	2.9410
48	5277968	29521	1553	58086	25669	7745	2748855	11979502	66493	425158	311910	15398932	2.9331
49	5277095	28955	1860	57009	27694	7601	2684364	11980516	65219	459800	306150	15365612	2.9268
50	5276159	28341	2182	56025	29717	7470	2626279	11981603	63837	494510	300904	15339459	2.9218
51	5275160	27863	2703	54878	31711	7317	2560865	11982775	62762	529939	294779	15305596	2.9154
52	5274069	27354	3286	53778	33691	7170	2499359	11984035	61620	565508	288900	15276182	2.9097
53	5272901	26800	3899	52724	35673	7030	2441762	11985380	60372	601297	283290	15251357	2.9050
54	5271663	26204	4541	51717	37664	6896	2388073	11986787	59031	637400	277927	15231156	2.9012
55	5270351	25563	5213	50756	39648	6767	2338292	11988291	57587	673574	272787	15215357	2.8982
56	5268947	25043	6096	49842	41724	6646	2292419	11989895	56417	712506	267939	15206342	2.8964
OPTIMUM:													
57	5267444	24548	7104	48974	43878	6530	2250455	11991618	55303	753445	263315	15203529	2.8959
58	5265842	24002	8160	48153	46089	6420	2212399	11993459	54076	795601	258938	15206321	2.8964
59	5264153	23413	9260	47378	48334	6317	2178251	11995419	52751	838571	254831	15214321	2.8980
60	5262376	22784	10408	46650	50623	6220	2148011	11997481	51333	882539	250970	15227669	2.9005
61	5260488	22239	11751	45861	52939	6115	2109274	11999626	50106	928120	246786	15233700	2.9017
62	5258491	21772	13281	45099	55279	6013	2072190	12001916	49056	975204	242736	15242990	2.9034
63	5256376	21251	14875	44364	57635	5915	2036759	12004350	47885	1022939	238841	15255004	2.9057
64	5254158	20687	16529	43656	59995	5821	2002981	12006932	46614	1071092	235102	15269493	2.9085
65	5251839	20085	18246	42975	62387	5730	1970856	12009669	45258	1120144	231496	15286906	2.9118
66	5249400	19526	20125	42321	64850	5643	1940383	12012557	43997	1171322	228046	15308311	2.9159
67	5246839	19088	22249	41694	67357	5559	1911563	12015513	43015	1224678	224728	15333467	2.9207
68	5244144	18592	24447	41094	69907	5479	1884395	12018648	41899	1279177	221567	15361889	2.9261
69	5241331	18052	26721	40521	72509	5403	1858882	12021940	40685	1334972	218562	15393670	2.9321
70	5238413	17475	29063	39975	75182	5330	1835020	12025418	39385	1392323	215690	15429066	2.9389
71	5235365	16898	31533	39402	77916	5254	1806150	12029066	38084	1451437	212688	15461258	2.9450
72	5232185	16478	34292	38847	80676	5180	1777721	12032928	37140	1512705	209775	15495989	2.9516
73	5228856	15992	37136	38309	83528	5108	1749735	12036879	36049	1575968	206948	15533481	2.9588
74	5225404	15463	40459	37788	86455	5038	1722189	12041004	34859	1640924	204209	15573467	2.9664
75	5221834	14835	43061	37284	89420	4971	1695086	12045298	33580	1706971	201580	15615356	2.9744
76	5218126	14288	46162	36798	92425	4906	1668424	12049788	32212	1774260	199039	15659299	2.9827
77	5214269	13862	49592	36329	95445	4844	1642204	12054518	31254	1843763	196609	15705841	2.9916
78	5210265	13374	53109	35877	98306	4784	1616425	12059507	30158	1911211	194267	15751252	3.0002
79	5206124	12841	56718	35442	101180	4726	1591089	12064623	28960	1979421	192013	15798185	3.0092
80	5201851	12268	60417	35025	104133	4670	1566193	12069865	27669	2049455	189846	15847691	3.0186

SECTION X

RESULTS OF THE ECONOMIC OPTIMIZATION

Economic optimization analyses of dry-type cooling systems for electric-generating plants were made for 27 selected sites in the United States, including one site each in the states of Hawaii and Alaska. The sites were selected to represent a range of annual air temperatures, ground-level elevation, and fuel cost, all of which have some effect on the economic optimization.

Four basic sets of economic optimization analyses were performed for each of the 27 sites. These basic analyses were for the following conditions:

1. Fossil-fueled generating plant, natural-draft tower.
2. Fossil-fueled generating plant, mechanical-draft tower.
3. Nuclear-fueled generating plant, natural-draft tower.
4. Nuclear-fueled generating plant, mechanical-draft tower.

Fifteen analyses were made for each site for each of the 4 basic conditions summarized above. These 15 analyses reflect the combination of 5 fixed-charge rates and 3 fuel cost assumptions. Therefore, a total of 4 times 15, or 60 analyses were made for each site reflecting the basic assumptions. In addition, as discussed in Section XI, some supplemental analyses were made to illustrate the effect of varying certain parameters over a wider range.

The basic assumptions used in the economic optimization analyses have been previously discussed in Section IX of this report.

Table 10 shows the 27 sites which were analyzed; summarizes the site air temperature conditions and ground elevations; and summarizes the assumptions made as to fuel cost and capital cost multipliers.

As discussed in Section IX of this report, the computer printouts show, for each analysis reflecting a specific set of assumptions, the total annual cost for those cost items which are affected by the selection of the dry cooling tower size, or ITD. As expected, the shape of the curve of annual cost versus ITD is affected by the assumption as to the season of peak electrical demand. Figure 41 shows typical curves of annual cost versus ITD for a summer peaking assumption and for a winter peaking assumption. Under the winter peaking assumption, the annual cost continually decreases with increasing ITD to the 80°F ITD limit established, reflecting

TABLE 10

Economic Optimization Analysis
Summary of Sites, Site Data and Study Assumptions (1)

Site No.	Area and City		Ground-Level Elevation (2) (ft.)	Ambient Air Temp. (°F)		Capital Cost Multiplier (4)	Fuel Cost Range (¢/10^6 Btu)		
				Annual Median	10 hrs. (3)		800-Mw Unit		
							Fossil	Nuclear	Peaking
1	Pacific:	Seattle, Wash.	400	50	91	1.0	25-40	11-20	40
2		San Francisco, Calif.	0	56	89	1.05	25-40	11-20	40
3		Los Angeles, Calif.	100	62	93	1.0	25-40	11-20	40
4	Mountain:	Great Falls, Mont.	3,700	46	94	1.0	15-30	11-20	40
5		Boise, Ida.	4,000	50	101	1.0	20-35	11-20	40
6		Casper, Wyo.	5,300	45	96	0.95	10-25	11-20	15-30
7		Reno, Nev.	4,400	48	101	1.05	25-40	11-20	40
8		Denver, Colo.	5,300	51	97	0.95	20-35	11-20	40
9		Phoenix, Ariz.	1,100	72	114	1.0	20-35	11-20	40
10	West North Central:	Bismarck, N. Dak.	1,600	43	100	1.0	12-25	11-20	40
11		Minneapolis, Minn.	800	47	97	1.0	25-40	11-20	40
12		Omaha, Neb.	1,000	53	103	1.0	25-40	11-20	40
13	West South Central:	Little Rock, Ark.	300	65	105	0.90	25-40	11-20	25-40
14		Midland, Tex.	2,900	65	105	0.95	20-35	11-20	20-35
15		New Orleans, La.	0	71	97	0.95/0.97	20-35	11-20	20-35
16	East North Central:	Green Bay, Wis.	700	44	92	1.0	25-40	11-20	40
17		Grand Rapids, Mich.	700	49	96	1.0	25-40	11-20	40
18		Detroit, Mich.	600	51	97	1.05	25-40	11-20	40
19		Chicago, Ill.	600	50	96	1.05	25-40	11-20	40
20	East South Central:	Nashville, Tenn.	600	64	103	0.90	18-30	11-20	40
21	New England:	Burlington, Vt.	300	46	92	0.95	25-40	11-20	40
22	Mid-Atlantic:	Philadelphia, Penna.	0	56	99	1.0	25-40	11-20	40
23	South-Atlantic:	Charleston, W. Va.	1,000	57	96	1.0	15-30	11-20	40
24		Atlanta, Ga.	1,000	64	100	0.95	25-40	11-20	40
25		Miami, Fla.	0	77	97	1.0/1.02	25-40	11-20	40
26	Hawaii:	Honolulu, Hawaii	0	76	92	1.10	30-45	11-20	40-45
27	Alaska:	Anchorage, Alas.	100	38	77	1.50	30-45	11-20	30-45

Note: See footnotes on following page.

TABLE 10 FOOTNOTES

(1) General assumptions:

a. Unit size: 800 mw.

b. The following basic conditions were studied for each site:

Fossil-fueled unit, natural-draft tower
Fossil-fueled unit, mechanical-draft tower
Nuclear-fueled unit, natural-draft tower
Nuclear-fueled unit, mechanical-draft tower

c. Fifteen analyses were made at each site for each of the 4 basic conditions. The 15 analyses reflect the combination of 3 fuel cost assumptions and 5 assumptions as to fixed-charge rates.

d. The fixed-charge rates assumed were 8%, 10%, 12%, 15%, and 18%.

e. A summer peak was assumed for all sites other than No. 27, Anchorage, Alaska.

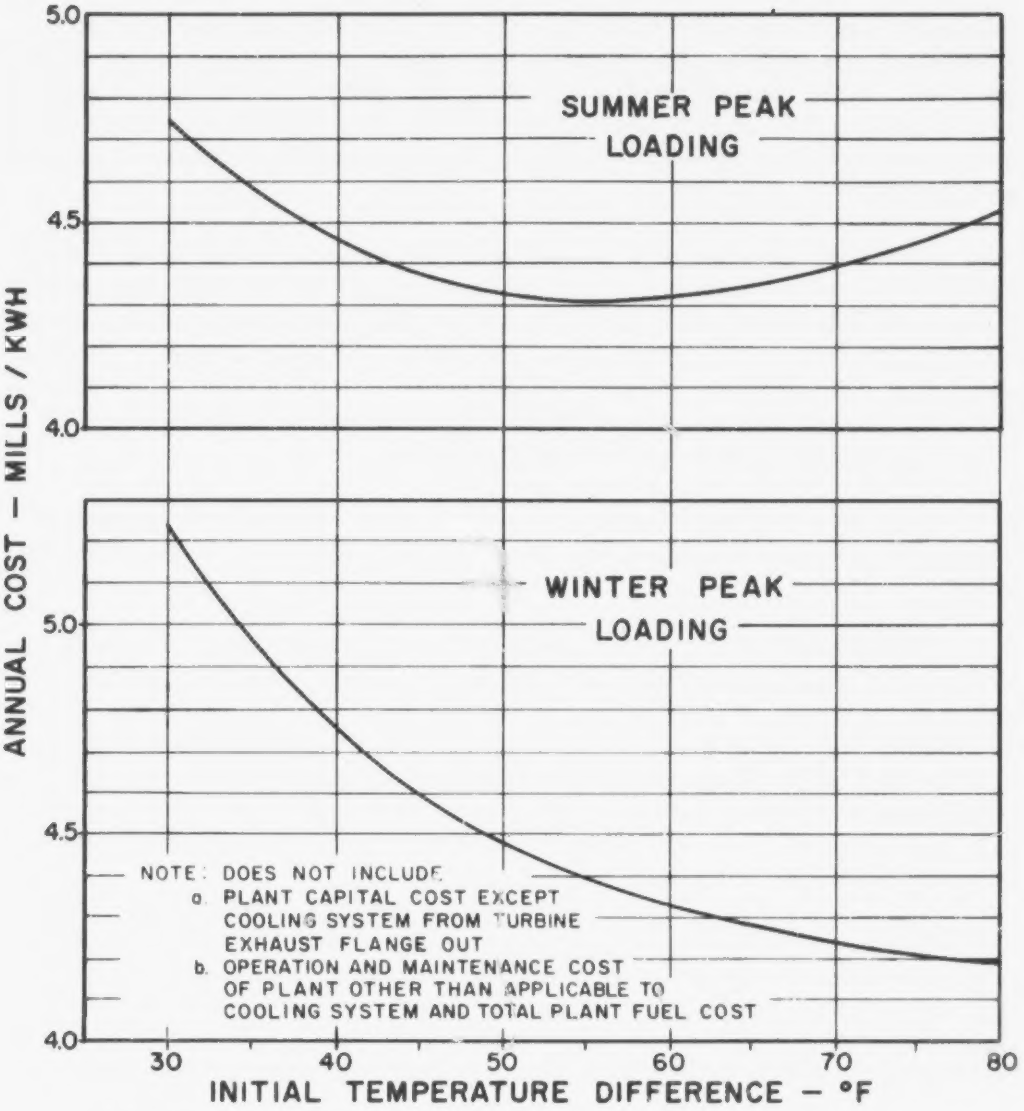
f. The capital cost of peaking capacity necessary to replace capacity lost at high back pressures was assumed to be $100/kw.

g. The incremental capital cost of the generating capacity necessary for cooling system pumps and fans was assumed to be $150/kw for fossil-fueled units and $225/kw for nuclear-fueled units. The auxiliary energy cost was assumed equal to the fuel cost of energy from the 800-mw unit.

(2) Weather station elevation rounded to the nearest 100 feet above sea level.

(3) The air temperature equalled or exceeded 10 hours per year.

(4) Reflects approximate construction cost differences and, for two sites, the additional cost of natural-draft towers in areas subject to hurricane winds. The lower multipliers shown for New Orleans and Miami are applicable to mechanical-draft cooling systems and the higher multipliers are applicable to natural-draft cooling systems.



LEGEND : 1. FUEL COST USED – 40 ¢ / 10^6 BTU
2. FIXED CHARGE RATE – 15%
3. CAPITAL COST MULTIPLIER
SUMMER CURVE – 1.0
WINTER CURVE – 1.5

FIGURE 41 – TYPICAL CURVES OF TOTAL ANNUAL COST (COOLING SYSTEM, PEAKING CAPACITY LOSS PENALTY AND TOTAL PLANT FUEL) VARIATION WITH ITD FOR SUMMER AND WINTER PEAKING ASSUMPTIONS

the capital cost versus ITD relationship. If this limit had not been arbitrarily established, the curve would eventually turn upward, indicating an optimized ITD selection. In the case of the summer peaking assumption, the annual cost declines with increasing ITD, up to a certain point, after which the economic effect of the assumptions as to the replacement of lost generating capability causes the curve to turn upward.

For those sites where a summer peak was assumed, it was found that the bottom of the optimization curve of annual cost versus ITD was fairly flat and that a range of ITD values could be defined for which the total annual cost of the cooling system was very close to the total annual cost at the optimum point. For the purposes of these analyses, the range of ITD values which are close to the optimum value has been defined as those values for which the total annual cost is within 0.01 mills per kwh of the cost at the optimum point.

The results of the economic optimization analyses reflecting the basic assumptions summarized in Table 10 are presented in the figures and tables described below.

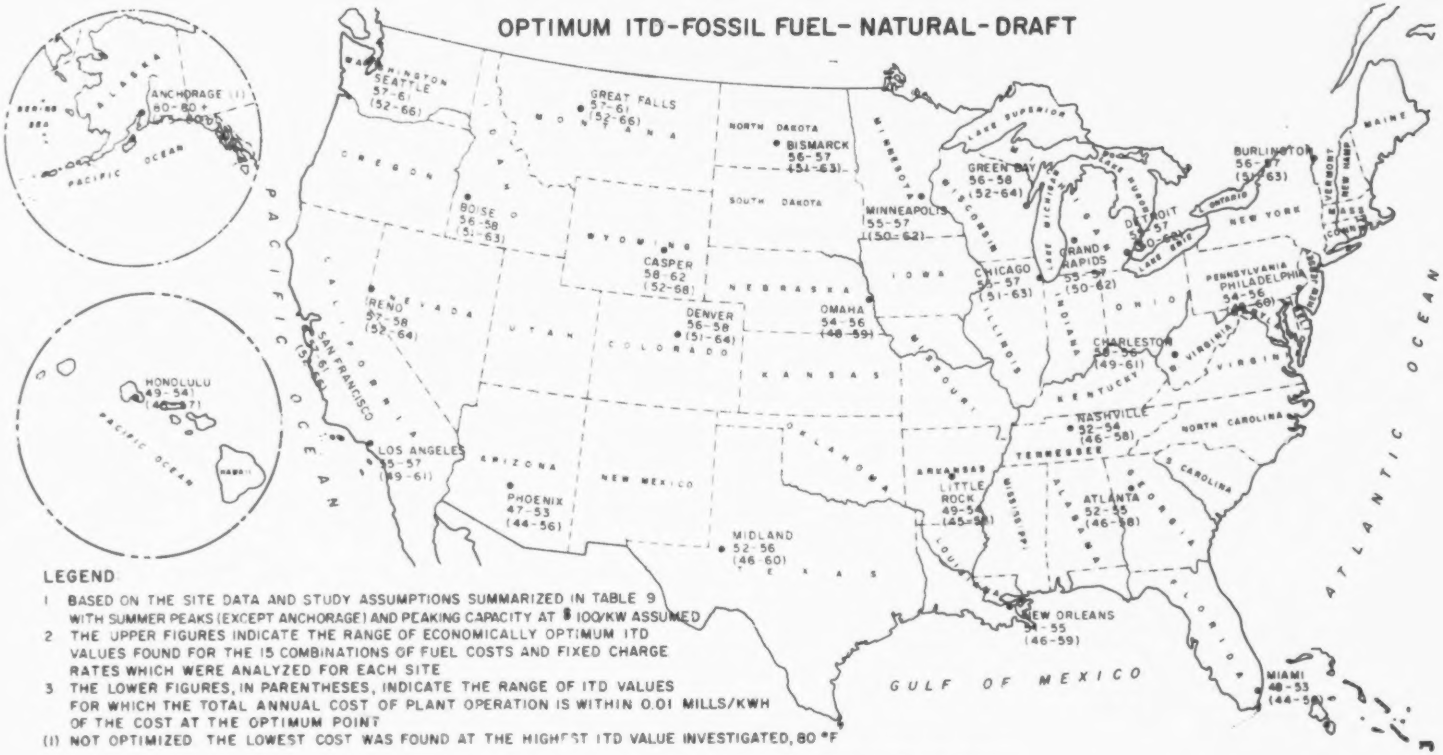
The economically optimum values of ITD are summarized on Figures 42 through 45. These figures show, on a map of the United States, the optimum ITD values found for the 15 combinations of fuel cost and fixed-charge rates which were investigated for each of the 27 sites. Figure 42 shows this information for the combination of a fossil-fueled generating unit and natural-draft tower. The other 3 basic sets of analyses—fossil-fueled unit, mechanical-draft tower; nuclear-fueled unit, natural-draft tower; and nuclear-fueled unit, mechanical-draft tower—are shown on Figures 43, 44 and 45, respectively.

Referring to Figure 42, it is noted that the range of economically optimum ITD values found for Chicago was 55°-57°F. The range of ITD values which were near the optimum (within 0.01 mills per kwh) was found to be 51°-63°F. In contrast, the range of economically optimum values for Miami was found to be 48°-53°F and the range of ITD values near the optimum was found to be 44°-56°F.

For Anchorage, Alaska where a winter peak was assumed, it was found that the total annual cost of the dry cooling system was lowest at the largest value of ITD investigated, 80°F, and, therefore, the dry cooling system was not optimized for the Anchorage site.

The results which are presented in this section will be discussed in detail in Section XI.

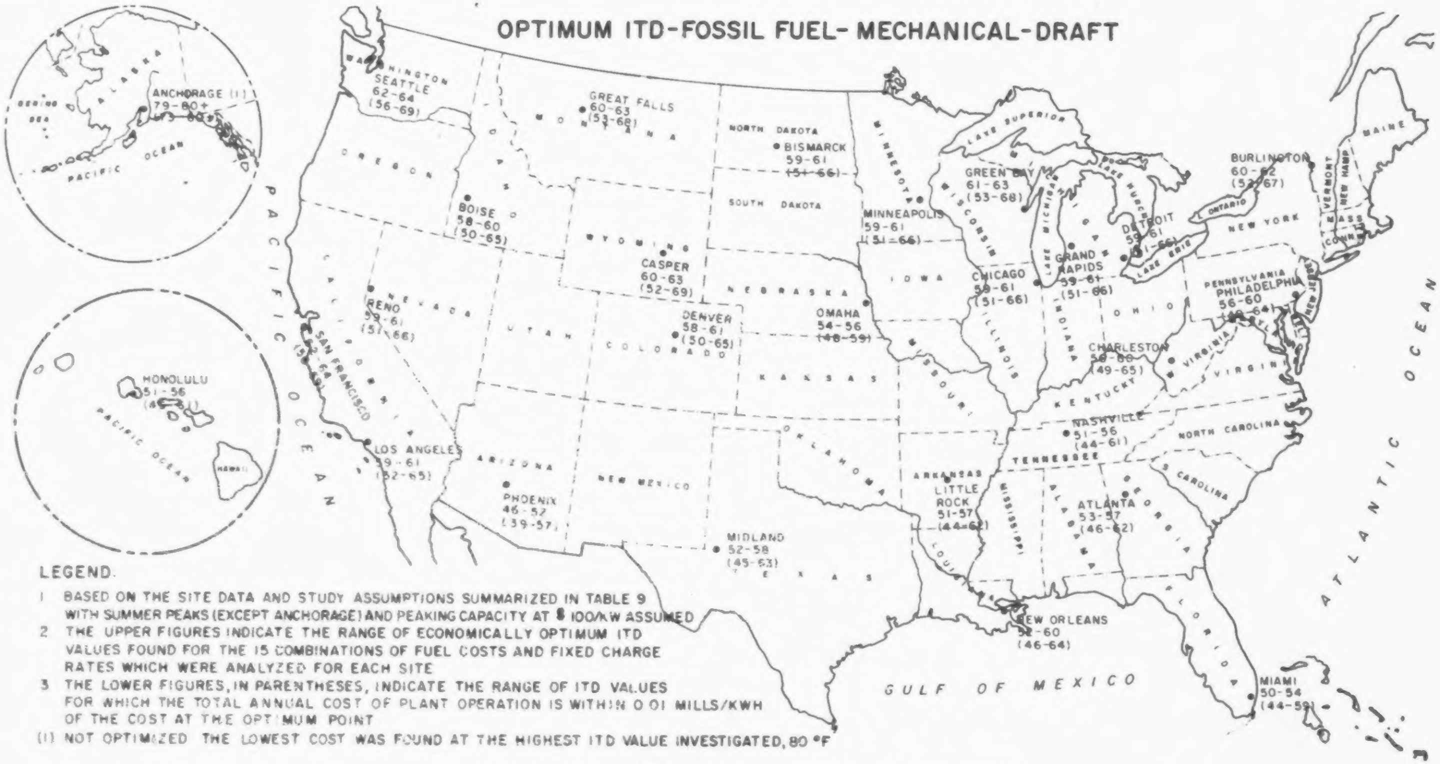
The generating capacity losses which would be experienced at the ambient air temperature equalled or exceeded 10 hours per year for the ITD values summa-


OPTIMUM ITD-FOSSIL FUEL- NATURAL- DRAFT

LEGEND

1 BASED ON THE SITE DATA AND STUDY ASSUMPTIONS SUMMARIZED IN TABLE 9 WITH SUMMER PEAKS (EXCEPT ANCHORAGE) AND PEAKING CAPACITY AT $ 100/KW ASSUMED
2 THE UPPER FIGURES INDICATE THE RANGE OF ECONOMICALLY OPTIMUM ITD VALUES FOUND FOR THE 15 COMBINATIONS OF FUEL COSTS AND FIXED CHARGE RATES WHICH WERE ANALYZED FOR EACH SITE
3 THE LOWER FIGURES, IN PARENTHESES, INDICATE THE RANGE OF ITD VALUES FOR WHICH THE TOTAL ANNUAL COST OF PLANT OPERATION IS WITHIN 0.01 MILLS/KWH OF THE COST AT THE OPTIMUM POINT
(1) NOT OPTIMIZED THE LOWEST COST WAS FOUND AT THE HIGHEST ITD VALUE INVESTIGATED, 80 °F

FIGURE 42 — ECONOMICALLY OPTIMUM VALUES OF INITIAL TEMPERATURE DIFFERENCE (°F) — FOSSIL-FUELED GENERATING UNIT — NATURAL-DRAFT TOWER



LEGEND.

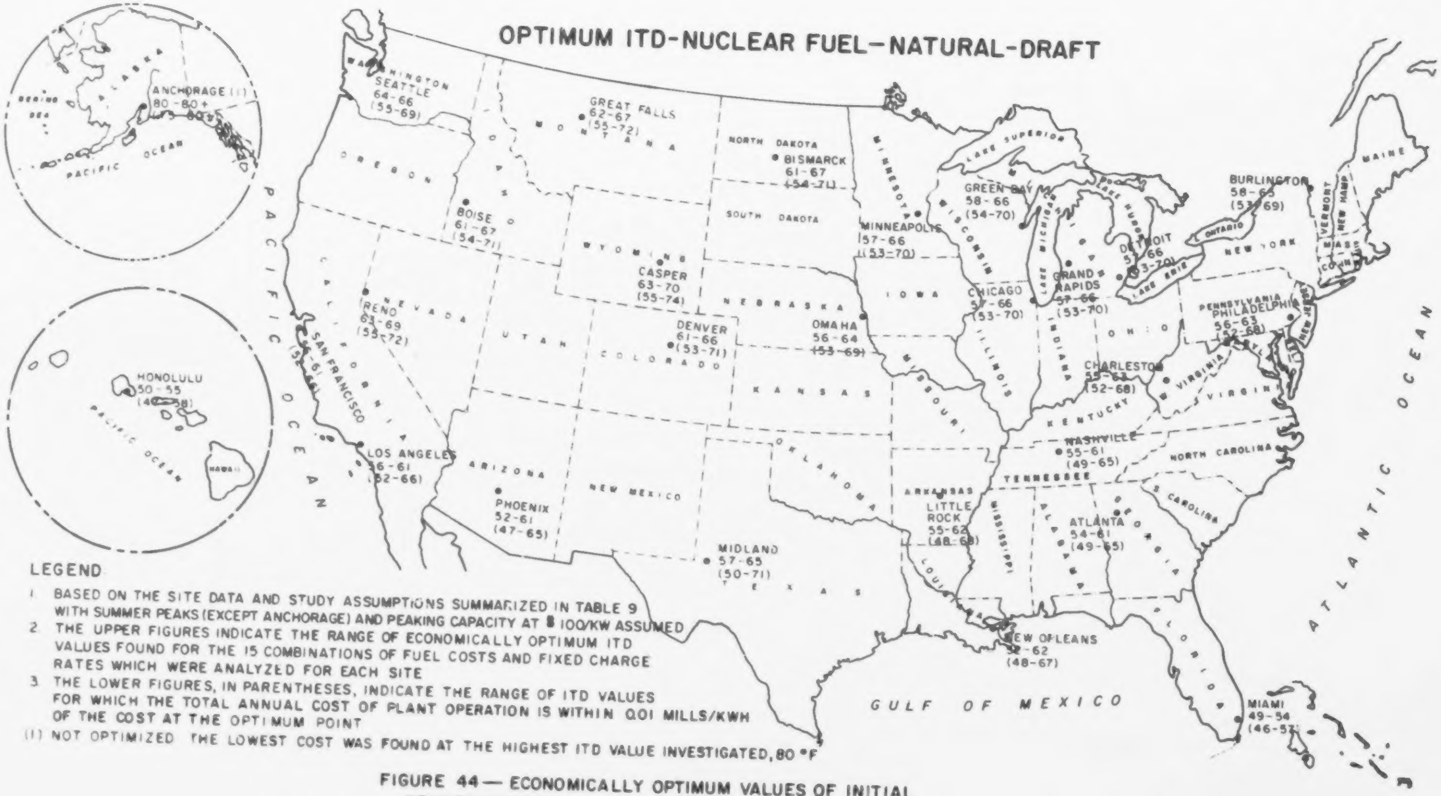
1 BASED ON THE SITE DATA AND STUDY ASSUMPTIONS SUMMARIZED IN TABLE 9 WITH SUMMER PEAKS (EXCEPT ANCHORAGE) AND PEAKING CAPACITY AT $ 100/KW ASSUMED

2 THE UPPER FIGURES INDICATE THE RANGE OF ECONOMICALLY OPTIMUM ITD VALUES FOUND FOR THE 15 COMBINATIONS OF FUEL COSTS AND FIXED CHARGE RATES WHICH WERE ANALYZED FOR EACH SITE

3 THE LOWER FIGURES, IN PARENTHESES, INDICATE THE RANGE OF ITD VALUES FOR WHICH THE TOTAL ANNUAL COST OF PLANT OPERATION IS WITHIN 0.01 MILLS/KWH OF THE COST AT THE OPTIMUM POINT

(1) NOT OPTIMIZED THE LOWEST COST WAS FOUND AT THE HIGHEST ITD VALUE INVESTIGATED, 80 °F

FIGURE 43 — ECONOMICALLY OPTIMUM VALUES OF INITIAL TEMPERATURE DIFFERENCE (°F) — FOSSIL-FUELED GENERATING UNIT— MECHANICAL-DRAFT TOWER



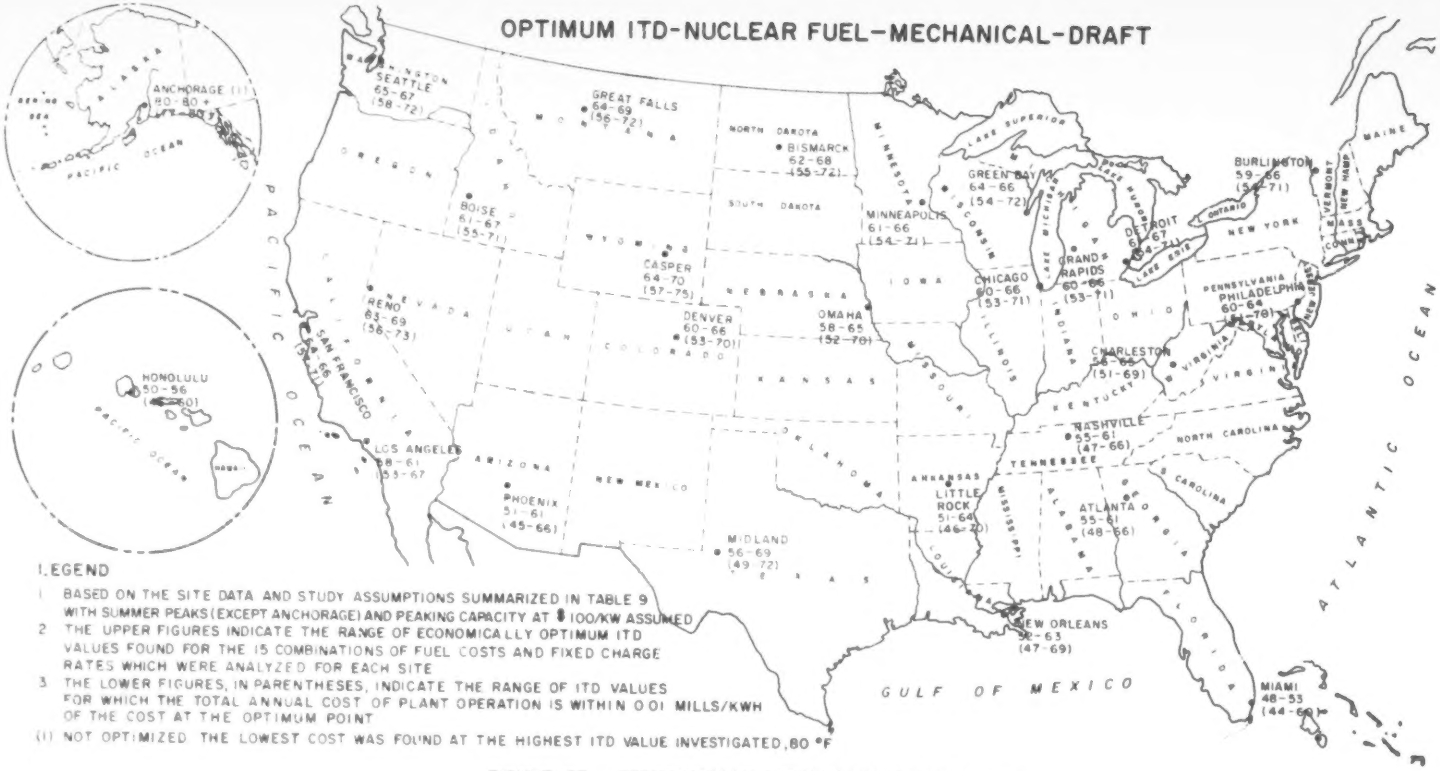
OPTIMUM ITD-NUCLEAR FUEL-NATURAL-DRAFT

LEGEND

1. BASED ON THE SITE DATA AND STUDY ASSUMPTIONS SUMMARIZED IN TABLE 9 WITH SUMMER PEAKS (EXCEPT ANCHORAGE) AND PEAKING CAPACITY AT $ 100/KW ASSUMED
2. THE UPPER FIGURES INDICATE THE RANGE OF ECONOMICALLY OPTIMUM ITD VALUES FOUND FOR THE 15 COMBINATIONS OF FUEL COSTS AND FIXED CHARGE RATES WHICH WERE ANALYZED FOR EACH SITE
3. THE LOWER FIGURES, IN PARENTHESES, INDICATE THE RANGE OF ITD VALUES FOR WHICH THE TOTAL ANNUAL COST OF PLANT OPERATION IS WITHIN 0.01 MILLS/KWH OF THE COST AT THE OPTIMUM POINT

(1) NOT OPTIMIZED. THE LOWEST COST WAS FOUND AT THE HIGHEST ITD VALUE INVESTIGATED, 80 °F

FIGURE 44 — ECONOMICALLY OPTIMUM VALUES OF INITIAL TEMPERATURE DIFFERENCE (°F) — NUCLEAR-FUELED GENERATING UNIT — NATURAL-DRAFT TOWER



LEGEND

1 BASED ON THE SITE DATA AND STUDY ASSUMPTIONS SUMMARIZED IN TABLE 9 WITH SUMMER PEAKS (EXCEPT ANCHORAGE) AND PEAKING CAPACITY AT $ 100/KW ASSUMED

2 THE UPPER FIGURES INDICATE THE RANGE OF ECONOMICALLY OPTIMUM ITD VALUES FOUND FOR THE 15 COMBINATIONS OF FUEL COSTS AND FIXED CHARGE RATES WHICH WERE ANALYZED FOR EACH SITE

3 THE LOWER FIGURES, IN PARENTHESES, INDICATE THE RANGE OF ITD VALUES FOR WHICH THE TOTAL ANNUAL COST OF PLANT OPERATION IS WITHIN 0.01 MILLS/KWH OF THE COST AT THE OPTIMUM POINT

(1) NOT OPTIMIZED. THE LOWEST COST WAS FOUND AT THE HIGHEST ITD VALUE INVESTIGATED, 80 °F

FIGURE 45 — ECONOMICALLY OPTIMUM VALUES OF INITIAL TEMPERATURE DIFFERENCE (°F) — NUCLEAR-FUELED GENERATING UNIT — MECHANICAL-DRAFT TOWER

rized in Figures 42 through 45 are also summarized on United States base maps, and these figures are designated as Figures 46 through 49.

To illustrate, Figure 46 shows that for Chicago the loss of capacity for the range of economically optimum ITD values when using a natural-draft cooling tower with a fossil-fueled generating unit would be on the order of 5.0 to 8.5 percent of the rated generating capacity.

Figures 50 through 53 summarize the capital cost of the dry cooling system for the range of the economically optimum ITD values. Again to illustrate, Figure 50 shows this range of capital cost, when using a natural-draft cooling tower with a fossil-fueled generating unit, for Chicago to be $17.6 to $22.2 per kw.

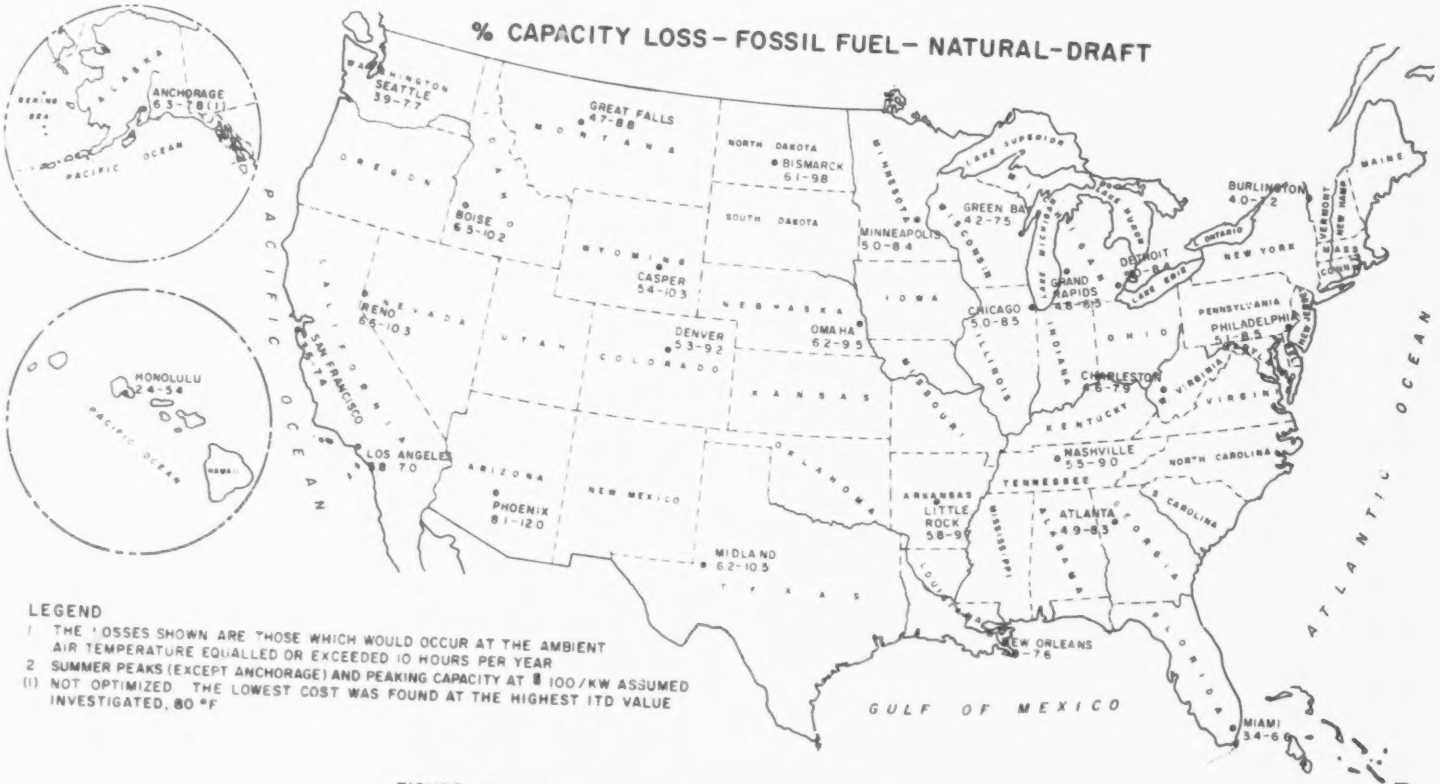
Figures 54 through 57 indicate the sum of the total capital cost of the dry cooling system and the capital cost of the required peaking capacity. The peaking capacity cost is applied to the total capacity of the 800,000-kw plant in order to determine the penalty per kw. The capital cost of the peaking capacity has been evaluated at $100 per kw of peaking capacity required. Figure 54 therefore shows that the combined cost of the dry cooling system and peaking capacity is $26.1 to $27.3 per kw for Chicago when using a natural-draft cooling tower with a fossil-fueled generating unit.

Much of the information shown on the United States base maps has also been summarized in Tables 11 through 22, with all dollar values per kw rounded to the nearest whole dollar. The information has been tabulated by fixed-charge rate in order to illustrate the effect of the fixed-charge rate on the economic optimization.

The economically optimum values of ITD are shown in Tables 11 through 14. For each site, 3 fossil-fuel costs and 3 nuclear-fuel costs were assumed. In many cases, for a given fixed-charge rate, the fuel cost variation did not have sufficient effect on the economic optimization to change the optimum ITD by a full degree F. In some cases, however, the fuel cost did affect the optimum and this is indicated in the tables. For example, as shown in Table 11, the value of the economically optimum ITD at Seattle for a 10 percent fixed-charge rate varied from 57° to 58°F for the range of fuel costs analyzed when using a natural-draft cooling tower with a fossil-fueled generating unit.

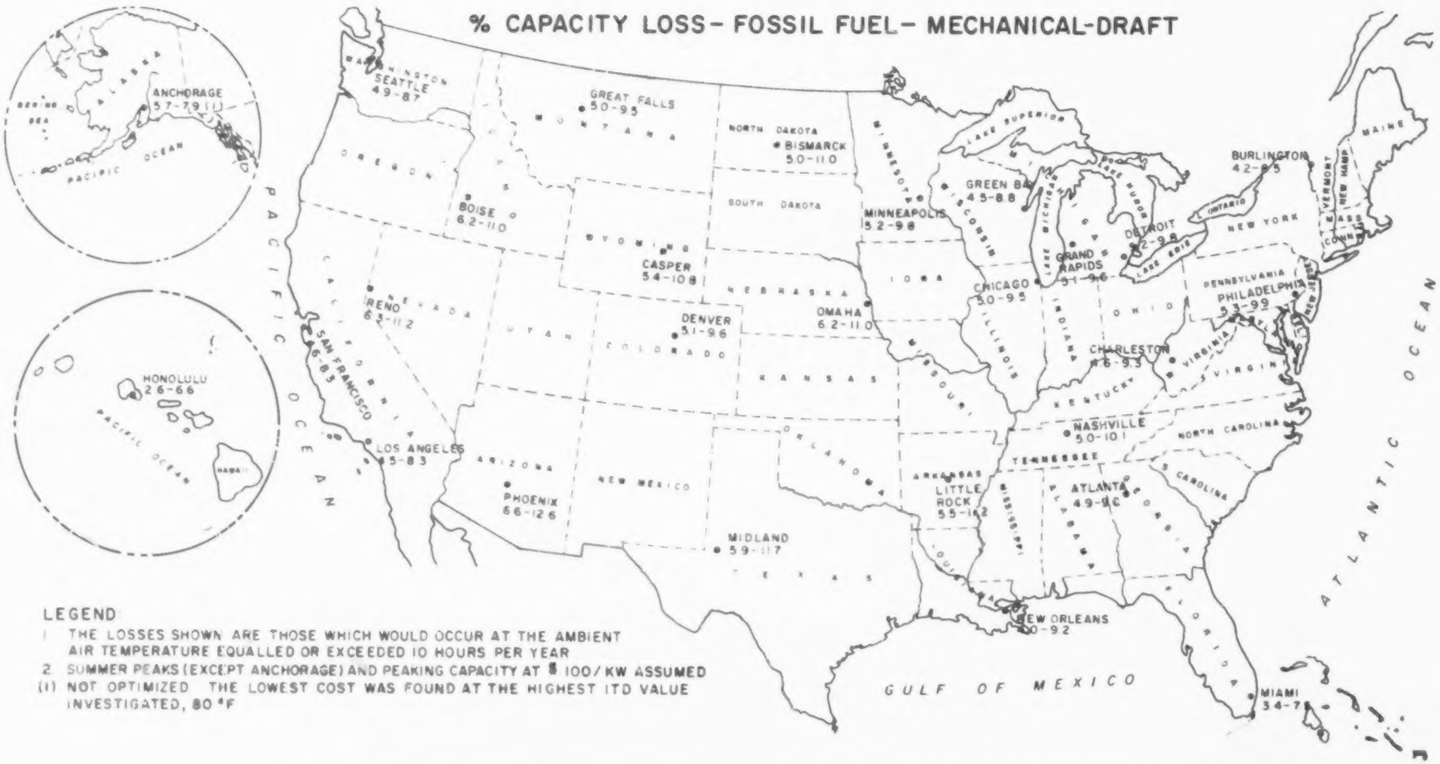
The capital cost of the dry cooling system is tabulated in Tables 15 through 18 for the range of optimum ITD values.

Tables 19 through 22 show the combined capital cost of the dry cooling system and the required peaking capacity.



LEGEND
1 THE LOSSES SHOWN ARE THOSE WHICH WOULD OCCUR AT THE AMBIENT AIR TEMPERATURE EQUALLED OR EXCEEDED 10 HOURS PER YEAR
2 SUMMER PEAKS (EXCEPT ANCHORAGE) AND PEAKING CAPACITY AT $ 100/KW ASSUMED
(1) NOT OPTIMIZED. THE LOWEST COST WAS FOUND AT THE HIGHEST ITD VALUE INVESTIGATED, 80 °F

FIGURE 46— GENERATING CAPACITY LOSSES AS PERCENT OF RATED LOAD FOR THE RANGE OF ECONOMICALLY OPTIMUM VALUES OF ITD SHOWN ON FIGURE 42— FOSSIL-FUELED GENERATING UNIT— NATURAL-DRAFT TOWER

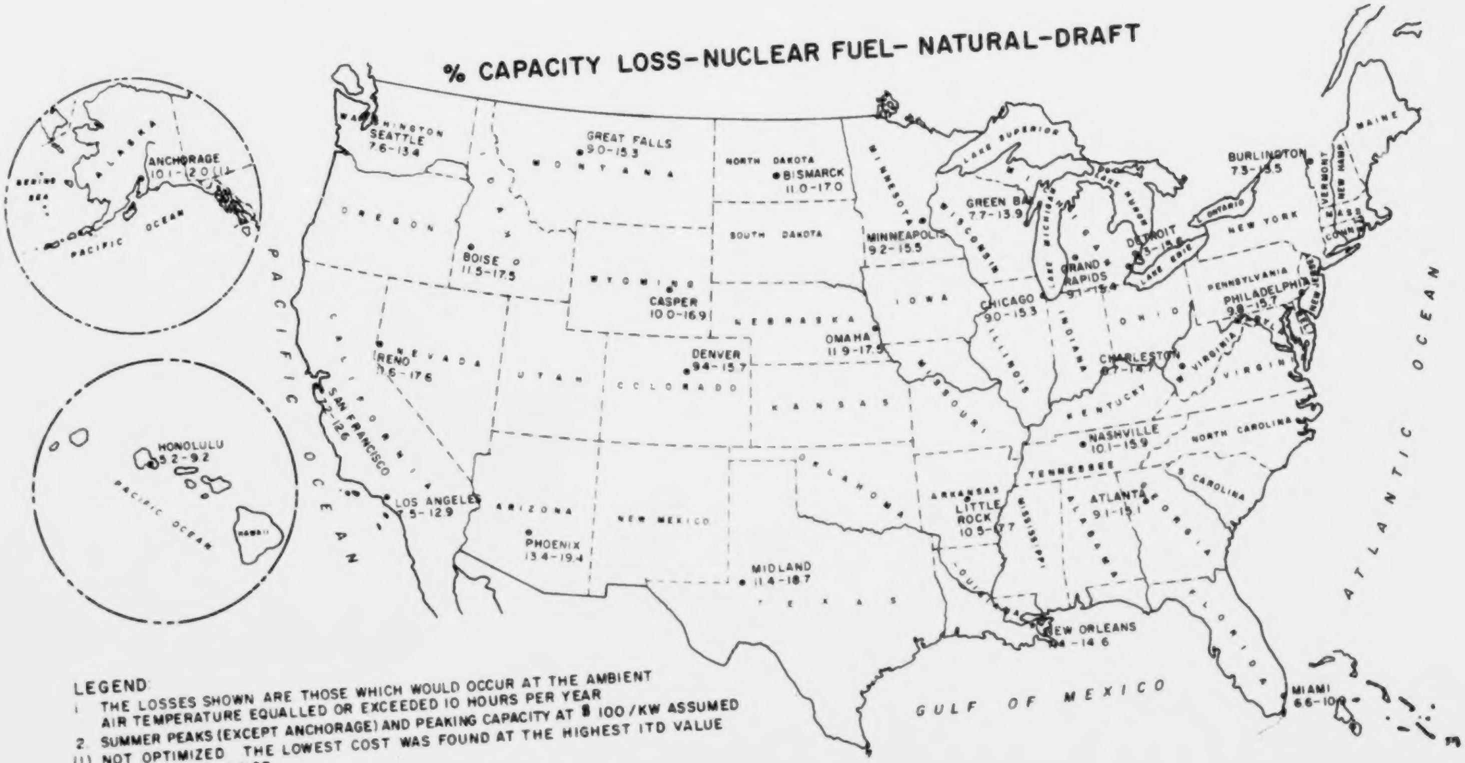


% CAPACITY LOSS- FOSSIL FUEL- MECHANICAL-DRAFT

LEGEND:
1. THE LOSSES SHOWN ARE THOSE WHICH WOULD OCCUR AT THE AMBIENT AIR TEMPERATURE EQUALLED OR EXCEEDED 10 HOURS PER YEAR
2. SUMMER PEAKS (EXCEPT ANCHORAGE) AND PEAKING CAPACITY AT $ 100/KW ASSUMED
(1) NOT OPTIMIZED. THE LOWEST COST WAS FOUND AT THE HIGHEST ITD VALUE INVESTIGATED, 80°F

FIGURE 47 — GENERATING CAPACITY LOSSES AS PERCENT OF RATED LOAD FOR THE RANGE OF ECONOMICALLY OPTIMUM VALUES OF ITD SHOWN ON FIGURE 43 FOSSIL-FUELED GENERATING UNIT— MECHANICAL-DRAFT TOWER

% CAPACITY LOSS-NUCLEAR FUEL- NATURAL-DRAFT

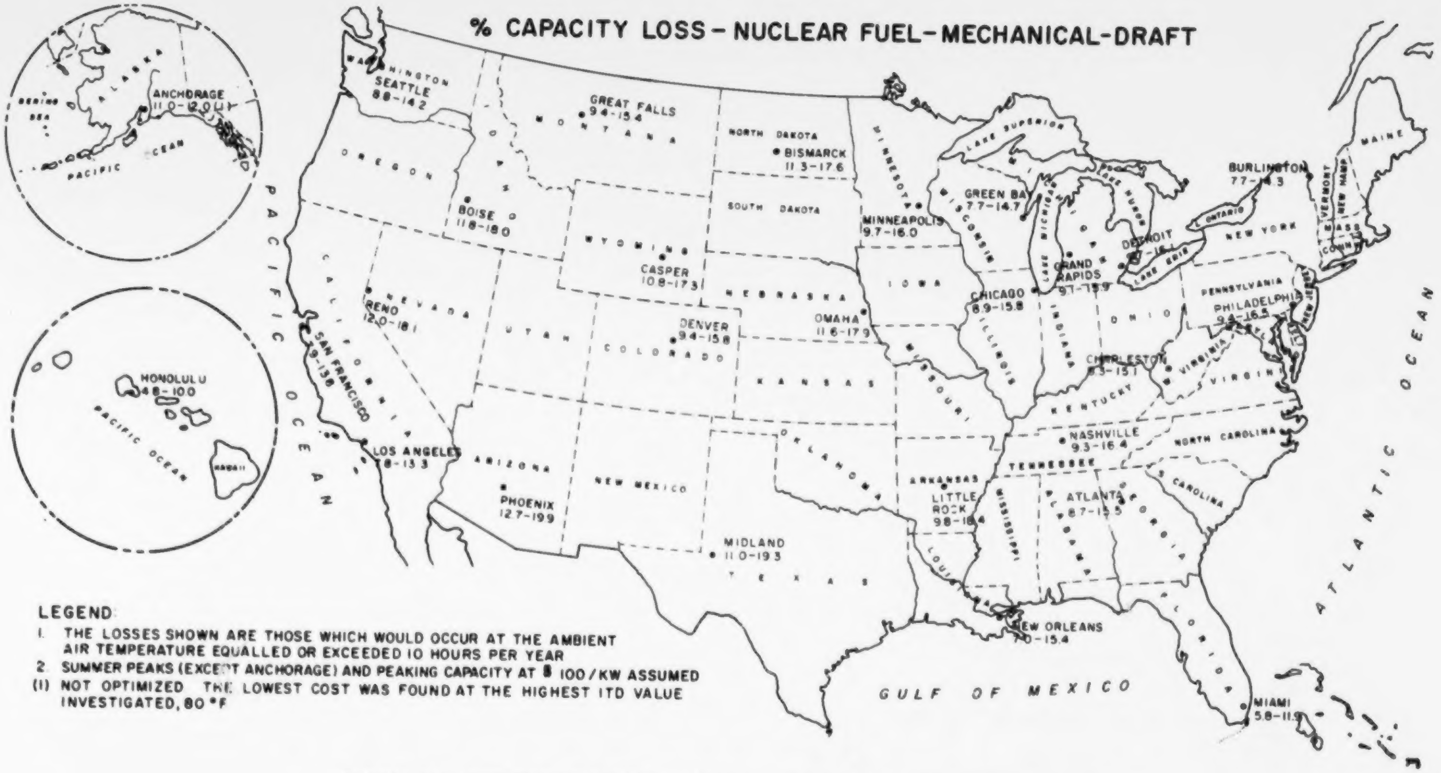


LEGEND

1. THE LOSSES SHOWN ARE THOSE WHICH WOULD OCCUR AT THE AMBIENT AIR TEMPERATURE EQUALLED OR EXCEEDED 10 HOURS PER YEAR
2. SUMMER PEAKS (EXCEPT ANCHORAGE) AND PEAKING CAPACITY AT $ 100/KW ASSUMED

(1) NOT OPTIMIZED THE LOWEST COST WAS FOUND AT THE HIGHEST ITD VALUE INVESTIGATED, 80 °F

FIGURE 48 — GENERATING CAPACITY LOSSES AS PERCENT OF RATED LOAD FOR THE RANGE OF ECONOMICALLY OPTIMUM VALUES OF ITD SHOWN ON FIGURE 44 NUCLEAR-FUELED GENERATING UNIT — NATURAL-DRAFT TOWER

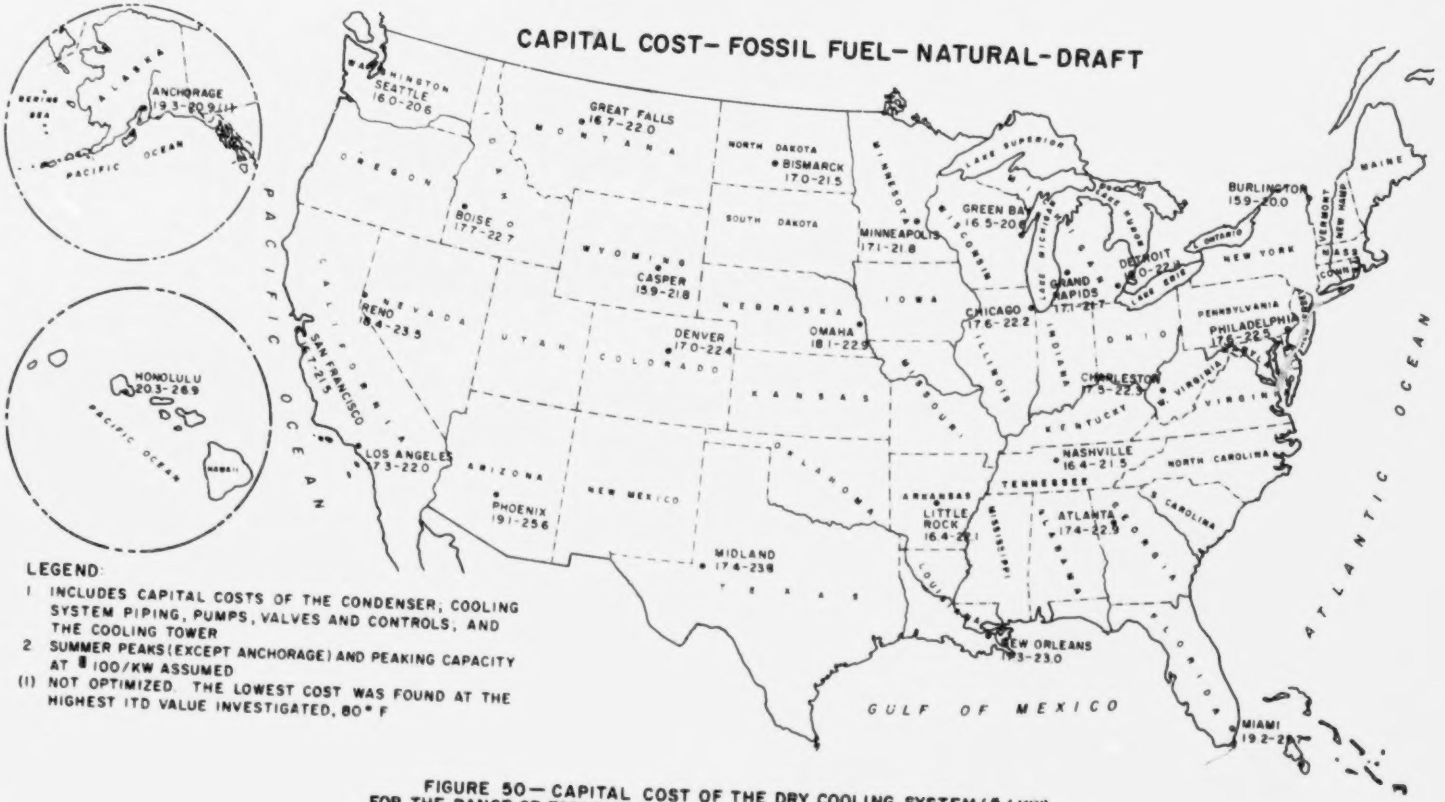


LEGEND:

1. THE LOSSES SHOWN ARE THOSE WHICH WOULD OCCUR AT THE AMBIENT AIR TEMPERATURE EQUALLED OR EXCEEDED 10 HOURS PER YEAR
2. SUMMER PEAKS (EXCEPT ANCHORAGE) AND PEAKING CAPACITY AT $ 100/KW ASSUMED

(1) NOT OPTIMIZED. THE LOWEST COST WAS FOUND AT THE HIGHEST ITD VALUE INVESTIGATED, 80 °F

FIGURE 49—GENERATING CAPACITY LOSSES AS PERCENT OF RATED LOAD FOR THE RANGE OF ECONOMICALLY OPTIMUM VALUES OF ITD SHOWN ON FIGURE 45 NUCLEAR-FUELED GENERATING UNIT—MECHANICAL-DRAFT TOWER



LEGEND:

1 INCLUDES CAPITAL COSTS OF THE CONDENSER; COOLING SYSTEM PIPING, PUMPS, VALVES AND CONTROLS; AND THE COOLING TOWER

2 SUMMER PEAKS (EXCEPT ANCHORAGE) AND PEAKING CAPACITY AT $100/KW ASSUMED

(1) NOT OPTIMIZED. THE LOWEST COST WAS FOUND AT THE HIGHEST ITD VALUE INVESTIGATED, 80° F

FIGURE 50— CAPITAL COST OF THE DRY COOLING SYSTEM ($/KW) FOR THE RANGE OF ECONOMICALLY OPTIMUM VALUES OF ITD SHOWN ON FIGURE 42 FOSSIL-FUELED GENERATING UNIT— NATURAL-DRAFT TOWER

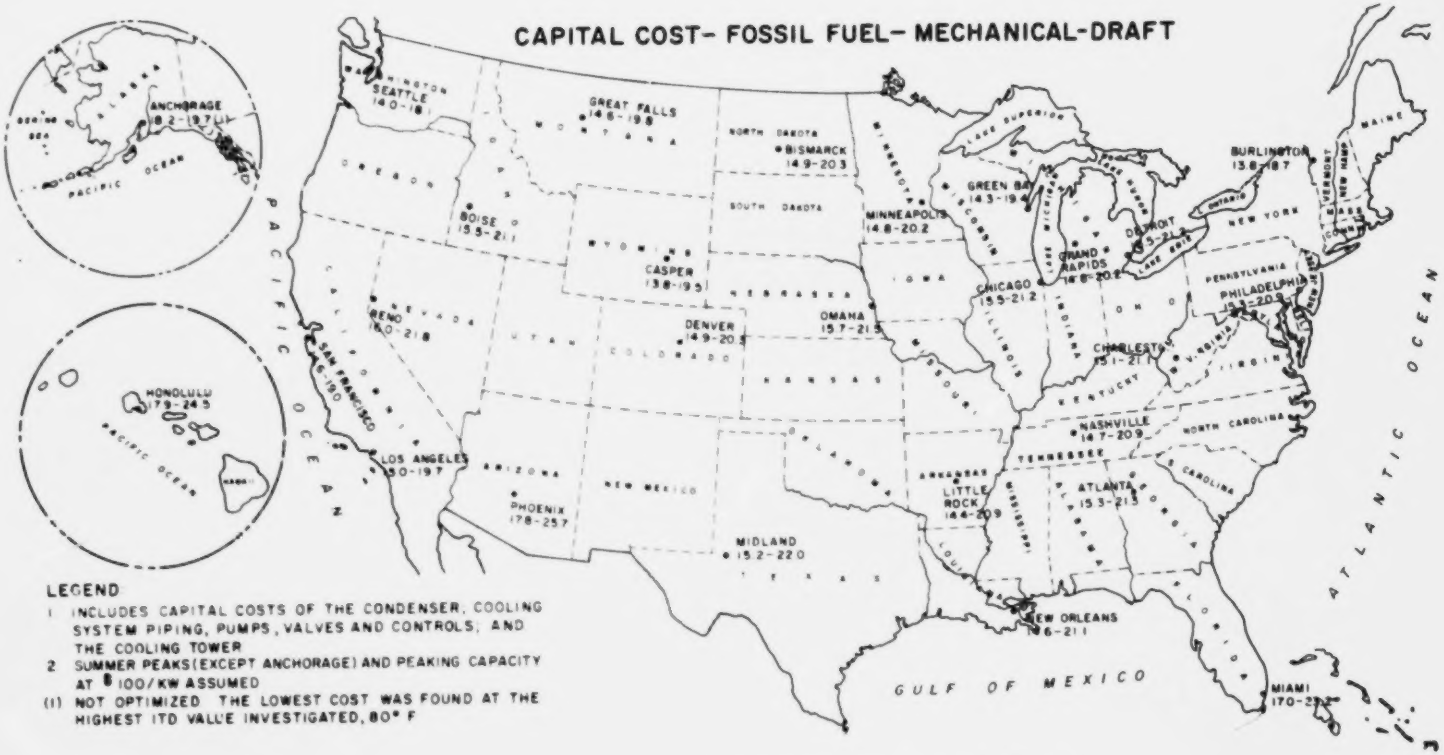
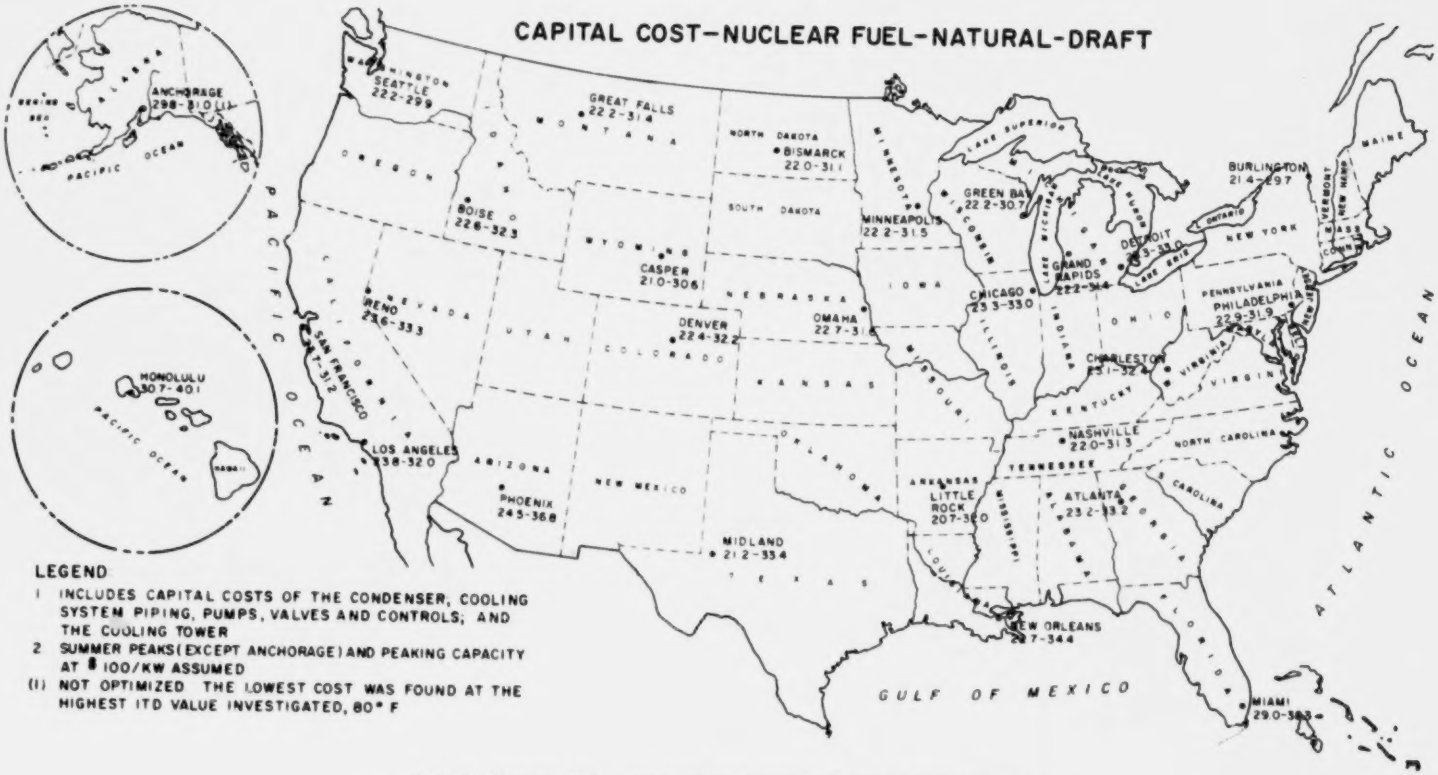


FIGURE 51 — CAPITAL COST OF THE DRY COOLING SYSTEM ($/KW) FOR THE RANGE OF ECONOMICALLY OPTIMUM VALUES OF ITD SHOWN ON FIGURE 43 FOSSIL-FUELED GENERATING UNIT — MECHANICAL- DRAFT TOWER

CAPITAL COST–NUCLEAR FUEL–NATURAL-DRAFT



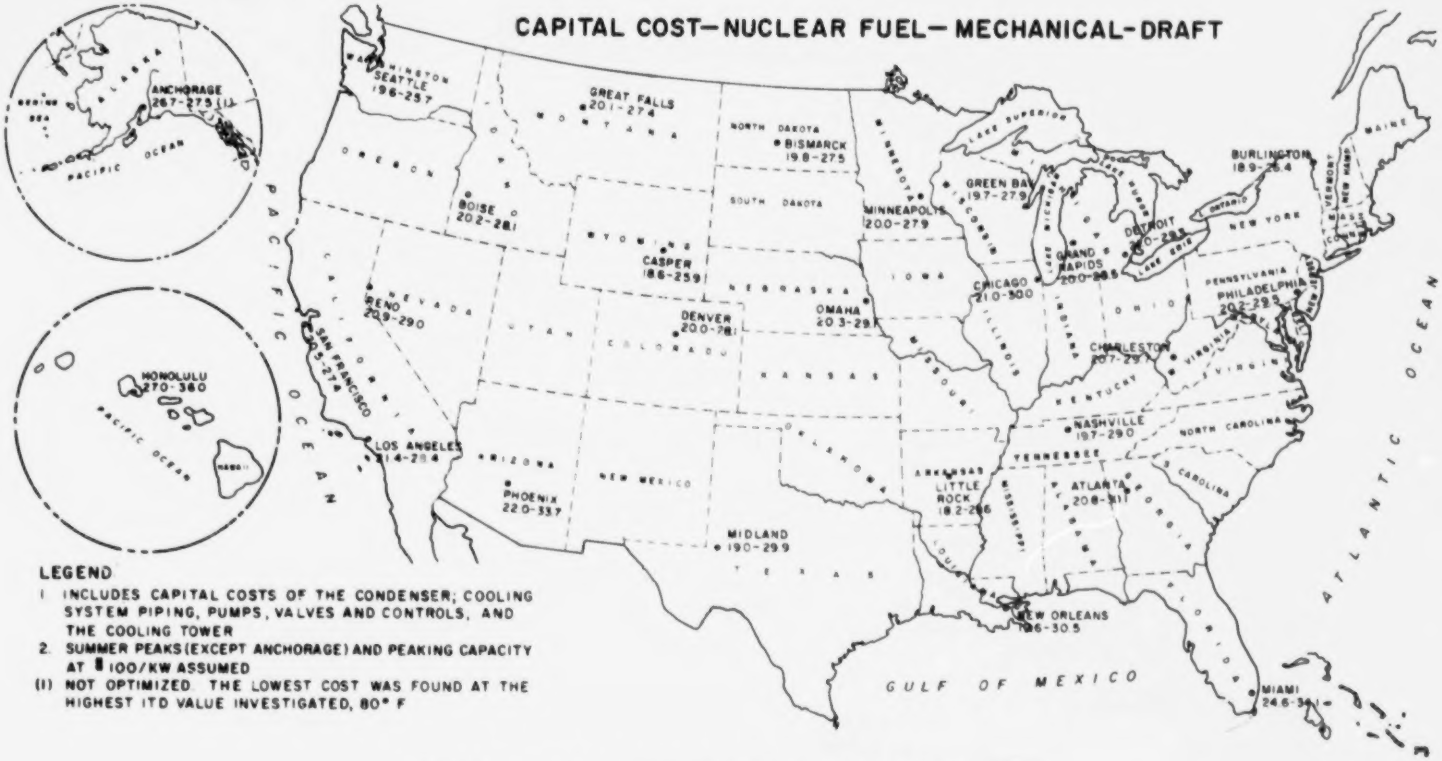
LEGEND

1 INCLUDES CAPITAL COSTS OF THE CONDENSER, COOLING SYSTEM PIPING, PUMPS, VALVES AND CONTROLS; AND THE COOLING TOWER

2 SUMMER PEAKS (EXCEPT ANCHORAGE) AND PEAKING CAPACITY AT $ 100/KW ASSUMED

(1) NOT OPTIMIZED. THE LOWEST COST WAS FOUND AT THE HIGHEST ITD VALUE INVESTIGATED, 80° F

FIGURE 52 — CAPITAL COST OF THE DRY COOLING SYSTEM ($/KW) FOR THE RANGE OF ECONOMICALLY OPTIMUM VALUES OF ITD SHOWN ON FIGURE 44 NUCLEAR-FUELED GENERATING UNIT — NATURAL-DRAFT TOWER

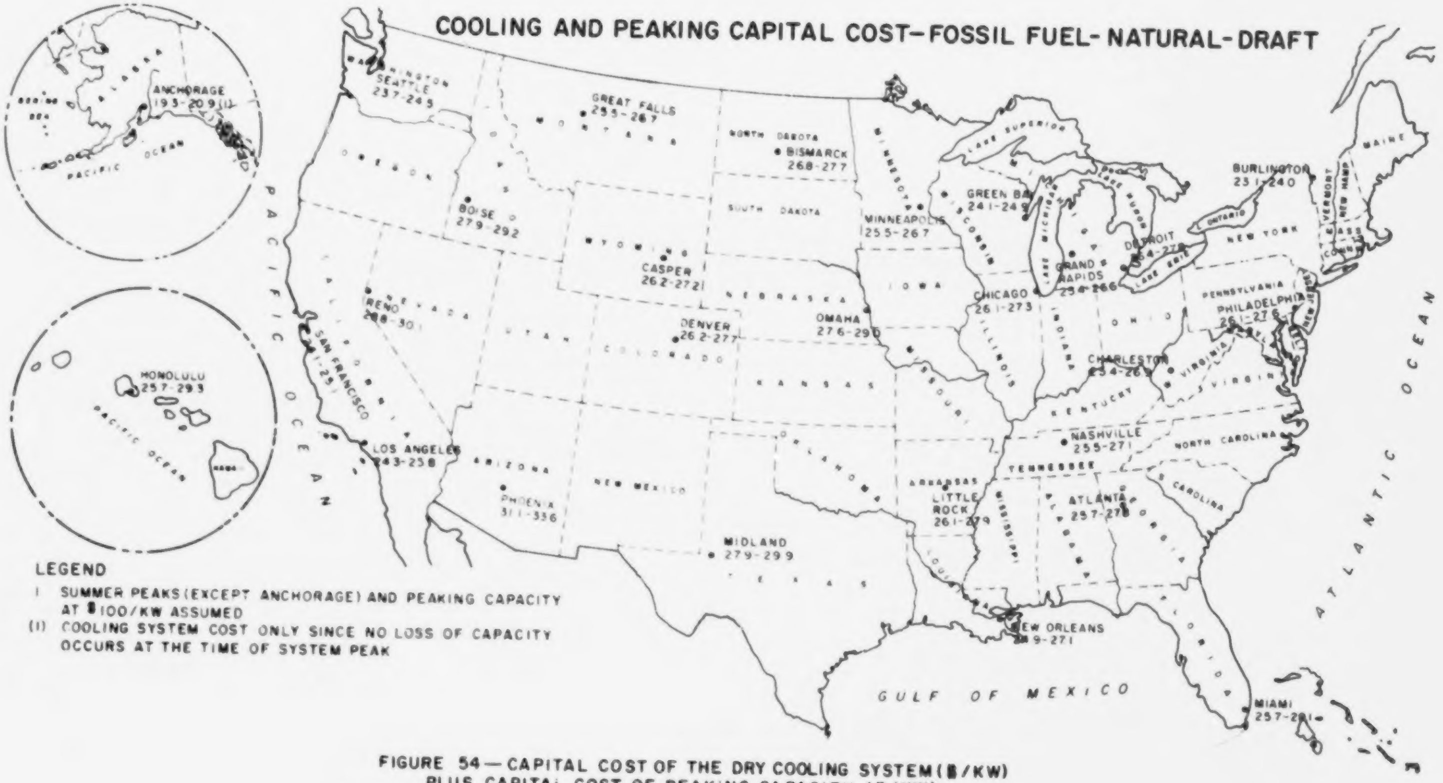


CAPITAL COST—NUCLEAR FUEL— MECHANICAL-DRAFT

LEGEND
1. INCLUDES CAPITAL COSTS OF THE CONDENSER; COOLING SYSTEM PIPING, PUMPS, VALVES AND CONTROLS, AND THE COOLING TOWER
2. SUMMER PEAKS (EXCEPT ANCHORAGE) AND PEAKING CAPACITY AT $ 100/KW ASSUMED

(1) NOT OPTIMIZED. THE LOWEST COST WAS FOUND AT THE HIGHEST ITD VALUE INVESTIGATED, 80° F

FIGURE 53 — CAPITAL COST OF THE DRY COOLING SYSTEM ($/KW)
FOR THE RANGE OF ECONOMICALLY OPTIMUM VALUES OF ITD SHOWN ON FIGURE 45
NUCLEAR-FUELED GENERATING UNIT— MECHANICAL-DRAFT TOWER



COOLING AND PEAKING CAPITAL COST–FOSSIL FUEL–NATURAL-DRAFT

LEGEND

I SUMMER PEAKS (EXCEPT ANCHORAGE) AND PEAKING CAPACITY AT $100/KW ASSUMED

(1) COOLING SYSTEM COST ONLY SINCE NO LOSS OF CAPACITY OCCURS AT THE TIME OF SYSTEM PEAK

FIGURE 54 — CAPITAL COST OF THE DRY COOLING SYSTEM ($/KW) PLUS CAPITAL COST OF PEAKING CAPACITY ($/KW) FOR THE RANGE OF ECONOMICALLY OPTIMUM VALUES OF ITD SHOWN ON FIGURE 42 FOSSIL-FUELED GENERATING UNIT— NATURAL-DRAFT TOWER

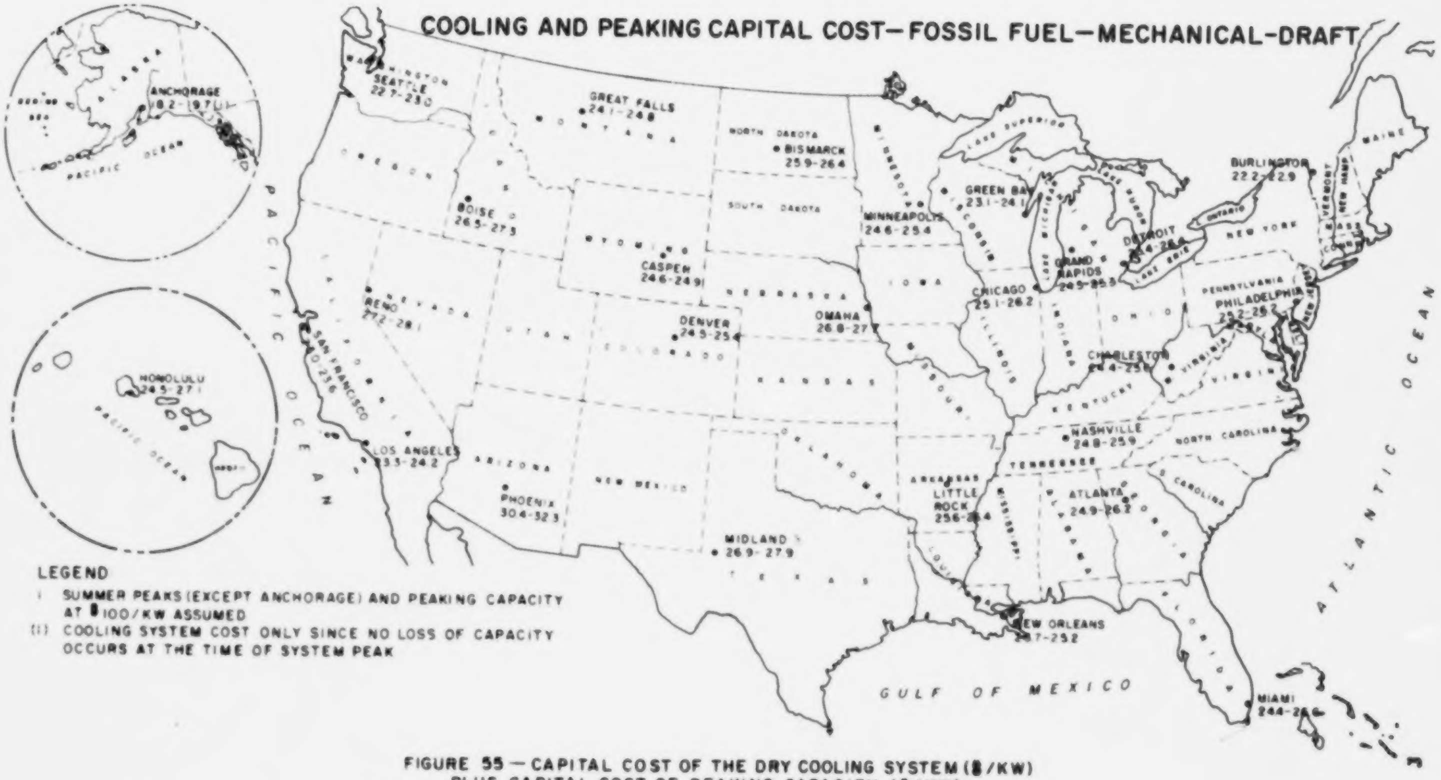
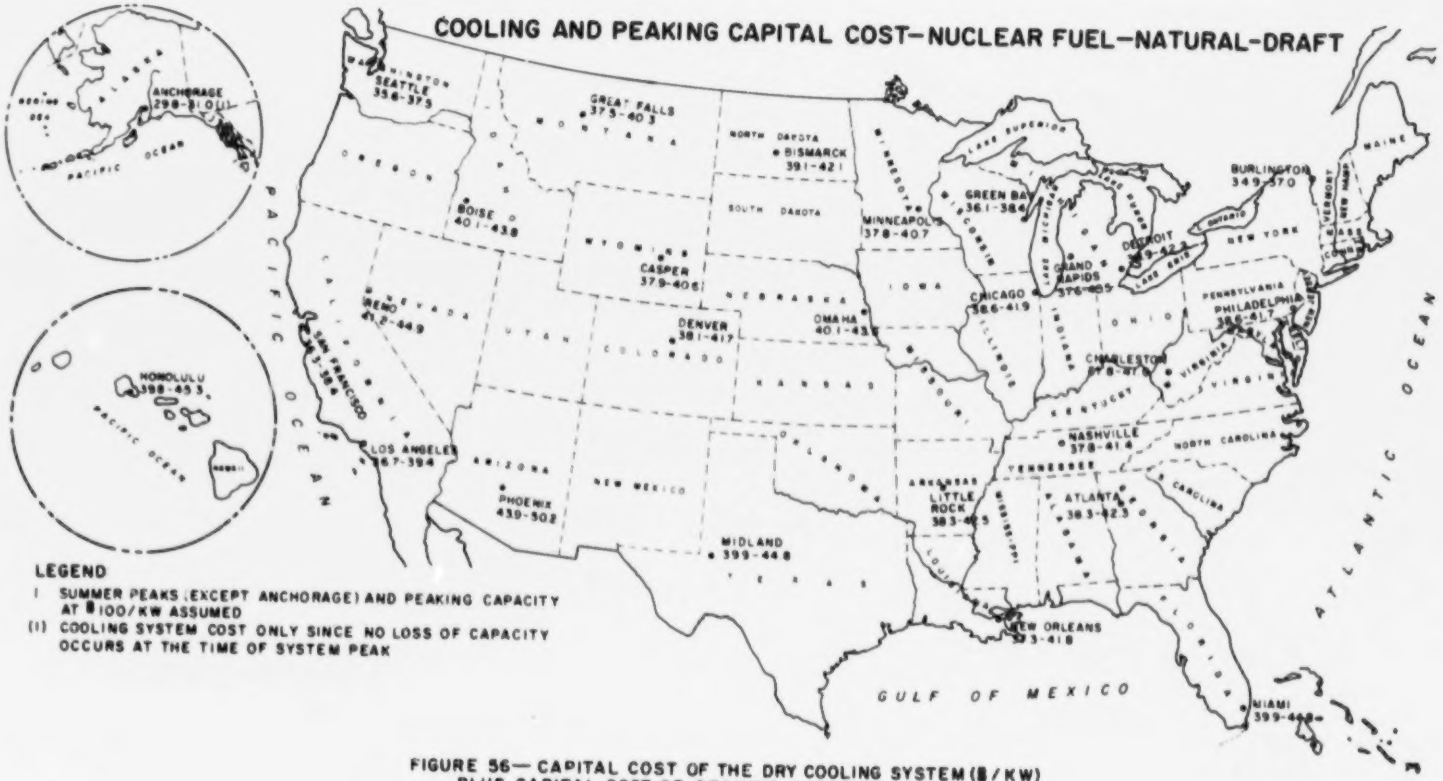


FIGURE 55 — CAPITAL COST OF THE DRY COOLING SYSTEM ($/KW)
PLUS CAPITAL COST OF PEAKING CAPACITY ($/KW)
FOR THE RANGE OF ECONOMICALLY OPTIMUM VALUES OF ITD SHOWN ON FIGURE 43
FOSSIL-FUELED GENERATING UNIT—MECHANICAL-DRAFT TOWER

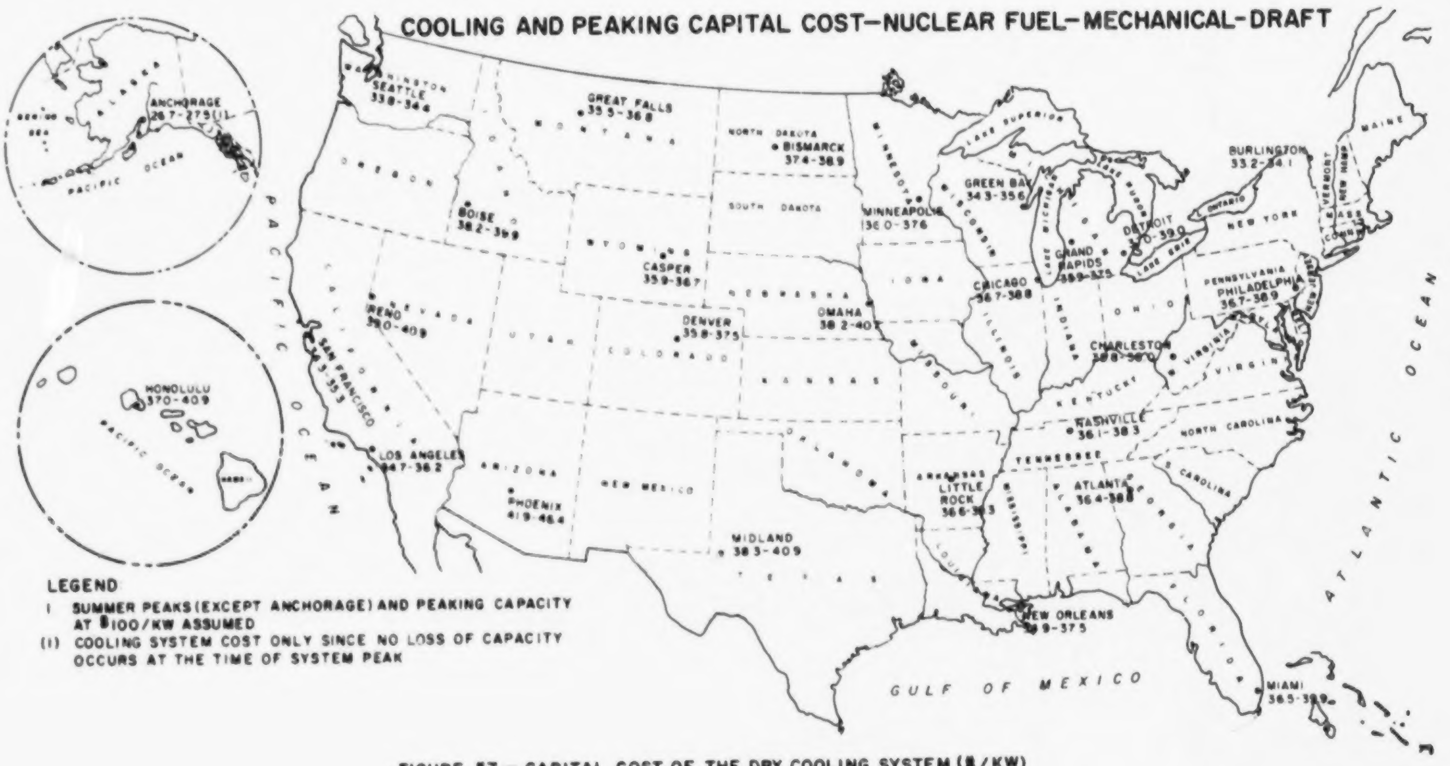

COOLING AND PEAKING CAPITAL COST—NUCLEAR FUEL—NATURAL-DRAFT

LEGEND

1 SUMMER PEAKS (EXCEPT ANCHORAGE) AND PEAKING CAPACITY AT $100/KW ASSUMED

(1) COOLING SYSTEM COST ONLY SINCE NO LOSS OF CAPACITY OCCURS AT THE TIME OF SYSTEM PEAK

FIGURE 56— CAPITAL COST OF THE DRY COOLING SYSTEM ($/KW)
PLUS CAPITAL COST OF PEAKING CAPACITY ($/KW)
FOR THE RANGE OF ECONOMICALLY OPTIMUM VALUES OF ITD SHOWN ON FIGURE 44
NUCLEAR-FUELED GENERATING UNIT— NATURAL-DRAFT TOWER


COOLING AND PEAKING CAPITAL COST—NUCLEAR FUEL—MECHANICAL-DRAFT

LEGEND

I SUMMER PEAKS (EXCEPT ANCHORAGE) AND PEAKING CAPACITY AT $100/KW ASSUMED

(1) COOLING SYSTEM COST ONLY SINCE NO LOSS OF CAPACITY OCCURS AT THE TIME OF SYSTEM PEAK

FIGURE 57 — CAPITAL COST OF THE DRY COOLING SYSTEM ($/KW)
PLUS CAPITAL COST OF PEAKING CAPACITY ($/KW)
FOR THE RANGE OF ECONOMICALLY OPTIMUM VALUES OF ITD SHOWN ON FIGURE 45
NUCLEAR-FUELED GENERATING UNIT—MECHANICAL-DRAFT TOWER

TABLE 11

Economically Optimum Values of Initial Temperature Difference (°F)
Fossil-Fueled Generating Unit, Natural-Draft Tower

	Initial Temperature Difference (°F)				
Fixed-Charge Rate:	8%	10%	12%	15%	18%
PLANT SITE					
Seattle, Wash.	57	57-58	58	58	58
San Francisco, Calif.	57	57-58	57-58	58	58
Los Angeles, Calif.	55	55-56	56	56	56-57
Great Falls, Mont.	57-58	58	58	59	59-61
Boise, Ida.	56	56-57	56-57	57	58
Casper, Wyo.	58-59	58-62	58-62	59-62	59-62
Reno, Nev.	57	57	58	58	58
Denver, Colo.	56	57	57	58	58
Phoenix, Ariz.	47-48	48	49	51-52	52-53
Bismarck, N. Dak.	56	56	57	57	57
Minneapolis, Minn.	55	56	56	57	57
Omaha, Neb.	54	55	55	55-56	56
Little Rock, Ark.	49-52	51-53	51-53	52-54	53-54
Midland, Tex.	52-54	53-55	54-55	54-56	55-56
New Orleans, La.	51-53	52-54	53-55	53-55	54-55
Green Bay, Wis.	57	57	57	57-58	58
Grand Rapids, Mich.	55-56	56	56	57	57
Detroit, Mich.	55-56	56	56	57	57
Chicago, Ill.	55-56	56	56	57	57
Nashville, Tenn.	52	52-53	53	54	54-55
Burlington, Vt.	56	56	57-58	57	57
Philadelphia, Penna.	54	54-55	55-56	55-56	56
Charleston, W. Va.	55	55	55-56	56	56-57
Atlanta, Ga.	52	53	54	54-55	55
Miami, Fla.	48-49	49-51	51	52	53
Honolulu, Hawaii	49	51-52	53-54	53	53-54
Anchorage, Alas. (1)	80	80	80	80	80

Note: Based upon the site data and study assumptions summarized in Table 10 with summer peaks (except Anchorage), peaking capacity at $100/kw and pump replacement capacity at $150/kw.

(1) Not optimized — winter peak assumed.

TABLE 12

Economically Optimum Values of Initial Temperature Difference (°F)
Fossil-Fueled Generating Unit, Mechanical-Draft Tower

	Initial Temperature Difference (°F)				
Fixed-Charge Rate:	8%	10%	12%	15%	18%
PLANT SITE					
Seattle, Wash.	62-63	63	64	64	64-65
San Francisco, Calif.	62	63	63	64	64-65
Los Angeles, Calif.	59	60	60	60	61
Great Falls, Mont.	60-61	61	62	63	63
Boise, Ida.	58	59	60	60	60
Casper, Wyo.	60-62	60-62	61-63	62-63	62-63
Reno, Nev.	59	60	60	61	61
Denver, Colo.	58	59	60	60	61
Phoenix, Ariz.	47	49	50	51	52
Bismarck, N. Dak.	59	60	60	60	61
Minneapolis, Minn.	59	60	60	60-61	61
Omaha, Neb.	55	56	57	58	59
Little Rock, Ark.	51-53	52-54	53-55	54-56	55-57
Midland, Tex.	52-55	53-56	54-57	55-58	61-63
New Orleans, La.	52-55	53-56	54-58	56-59	57-60
Green Bay, Wis.	60	61	61-62	62	63
Grand Rapids, Mich.	59	60	60	60-61	61
Detroit, Mich.	59	60	60	61	61
Chicago, Ill.	59	60	60	61	61
Nashville, Tenn.	51-52	53	54	55	56
Burlington, Vt.	60	60	61	61	62
Philadelphia, Penna.	56	57-58	59	60	60
Charleston, W. Va.	56	58	59	60	60
Atlanta, Ga.	53	55	55-56	56-57	57-58
Miami, Fla.	50	51	51-52	53	54
Honolulu, Hawaii	51	52-53	53-54	55	56
Anchorage, Alas. (1)	79-80	80	80	80	80

Note: Based upon the site data and study assumptions summarized in Table 10 with summer peaks (except Anchorage), peaking capacity at $100/kw; fan and pump replacement capacity at $150/kw.

(1) Not optimized — winter peak assumed.

TABLE 13

Economically Optimum Values of Initial Temperature Difference (^oF)
Nuclear-Fueled Generating Unit, Natural-Draft Tower

	Initial Temperature Difference (^oF)				
Fixed-Charge Rate:	8%	10%	12%	15%	18%
PLANT SITE					
Seattle, Wash.	59-65	65	65	65-66	66
San Francisco, Calif.	58-59	59-63	64-65	65	65
Los Angeles, Calif.	56	57	57-58	58	58-59
Great Falls, Mont.	63-65	65	66	66	67
Boise, Ida.	58-62	64	64-65	65	67
Casper, Wyo.	65-68	66-69	66-70	68-70	69-70
Reno, Nev.	63-64	64-65	65-66	67-68	68-69
Denver, Colo.	57-59	61-62	63	65-66	66
Phoenix, Ariz.	52-53	54-55	55	56	57-58
Bismarck, N. Dak.	61-62	63-64	64-65	65	65-67
Minneapolis, Minn.	57-58	61	62	63-65	65-66
Omaha, Neb.	56-57	57-58	58	63-64	64-65
Little Rock, Ark.	53-56	55-56	55-57	56-58	57-58
Midland, Tex.	55-58	56-58	57-64	58-65	58-66
New Orleans, La.	52-55	53-56	54-56	55-61	56-62
Green Bay, Wis.	58	58-59	65	65	66
Grand Rapids, Mich.	57-58	59-61	61-62	63-65	65-66
Detroit, Mich.	57-59	61	62	63-65	66
Chicago, Ill.	57-58	59-61	61-62	65	66
Nashville, Tenn.	55	55-56	56	56-57	57
Burlington, Vt.	58	58	58	65	65
Philadelphia, Penna.	56	56	57-58	61-62	62-63
Charleston, W. Va.	55-56	56-57	58	61	62-63
Atlanta, Ga.	54-55	55	56	56-57	57-58
Miami, Fla.	49	51	52-53	53	54
Honolulu, Hawaii	50-51	51-52	52-53	53-54	54-55
Anchorage, Alas. (1)	80	80	80	80	80

Note: Based upon the site data and study assumptions summarized in Table 10 with summer peaks (except Anchorage), peaking capacity at $100/kw and pump replacement capacity at $225/kw.

(1) Not optimized — winter peak assumed.

TABLE 14

Economically Optimum Values of Initial Temperature Difference (°F) Nuclear-Fueled Generating Unit, Mechanical-Draft Tower

	Initial Temperature Difference (°F)				
Fixed-Charge Rate:	8%	10%	12%	15%	18%
PLANT SITE					
Seattle, Wash.	65	66	66	67	67
San Francisco, Calif.	64	65	65	66	66
Los Angeles, Calif.	58-59	60	60	61	61-62
Great Falls, Mont.	65	65-66	66	67	69
Boise, Ida.	61	64	64	65	67
Casper, Wyo.	64-69	66-69	66-69	68-70	69-70
Reno, Nev.	63	64	65-66	68	69
Denver, Colo.	60-61	61	62	65	66
Phoenix, Ariz.	51	54	55	58-60	60-61
Bismarck, N. Dak.	62	64	64-65	67	68
Minneapolis, Minn.	61	61	63	66	66
Omaha, Neb.	58	61	62	64	65
Little Rock, Ark.	51-57	56-69	56-60	58-64	60-64
Midland, Tex.	56-60	57-63	59-64	60-65	63-66
New Orleans, La.	52-55	53-60	54-61	60-62	60-63
Green Bay, Wis.	61	65	65	66	66
Grand Rapids, Mich.	60-61	61	63	66	66
Detroit, Mich.	61	61-62	63	66	66-67
Chicago, Ill.	60	61	65	66	66-67
Nashville, Tenn.	55	56	57	59-60	61
Burlington, Vt.	59	61	65	65	66
Philadelphia, Penna.	60	61	61	63	64
Charleston, W. Va.	56-59	60	61	62	65
Atlanta, Ga.	55	56	57-60	60-61	61
Miami, Fla.	48	50	51	53	53
Honolulu, Hawaii	50	51	51-52	53	54-56
Anchorage, Alas. (1)	80	80	80	80	80

Note: Based upon the site data and study assumptions summarized in Table 10 with summer peaks (except Anchorage), peaking capacity at $100/kw; fan and pump replacement capacity at $225/kw.

(1) Not optimized — winter peak assumed.

TABLE 15

Capital Cost of the Dry Cooling System ($/Kw) for the Economically Optimum Values of Initial Temperature Difference Shown in Table 11 Fossil-Fueled Generating Unit, Natural-Draft Tower

	$/Kw				
Fixed-Charge Rate:	8%	10%	12%	15%	18%
PLANT SITE					
Seattle, Wash.	19	18-19	18	18	18
San Francisco, Calif.	19	19	19	19	19
Los Angeles, Calif.	19	19	19	19	18-19
Great Falls, Mont.	19-20	19	19	19	18-19
Boise, Ida.	20	20	20	20	19
Casper, Wyo.	19	18-19	18-19	18-19	17-19
Reno, Nev.	21	21	21	21	21
Denver, Colo.	20	19	19	19	19
Phoenix, Ariz.	23	23	22	21	20-21
Bismarck, N. Dak.	19	19	19	19	19
Minneapolis, Minn.	19	19	19	19	19
Omaha, Neb.	20	19	19	19	19
Little Rock, Ark.	19-20	18-19	18-19	18	18
Midland, Tex.	20-21	19-20	19-20	19-20	19
New Orleans, La.	19-20	19-20	19	19	19
Green Bay, Wis.	19	19	19	18-19	18
Grand Rapids, Mich.	19	19	19	19	19
Detroit, Mich.	20	20	20	20	20
Chicago, Ill.	20	20	20	20	20
Nashville, Tenn.	19	18-19	18	18	17-18
Burlington, Vt.	18	18	18	18	18
Philadelphia, Penna.	20	19-20	19	19	19
Charleston, W. Va.	19	19	19	19	19
Atlanta, Ga.	20	19	19	18-19	18
Miami, Fla.	22-23	21-22	21	21	20
Honolulu, Hawaii	24	23	23	22	22
Anchorage, Alas. (1)	19	19	19	19	19

Note: Based upon the site data and study assumptions summarized in Table 10 with summer peaks (except Anchorage), peaking capacity at $100/kw and pump replacement capacity at $150/kw.

(1) Not optimized — winter peak assumed.

TABLE 16

Capital Cost of the Dry Cooling System ($/Kw) for the Economically Optimum Values of Initial Temperature Difference Shown in Table 12 Fossil-Fueled Generating Unit, Mechanical-Draft Tower

	$/Kw				
Fixed-Charge Rate:	8%	10%	12%	15%	18%
PLANT SITE					
Seattle, Wash.	16	16	15	15	15
San Francisco, Calif.	17	16	16	16	16
Los Angeles, Calif.	17	17	17	17	16
Great Falls, Mont.	17	17	16	16	16
Boise, Ida.	18	17	17	17	17
Casper, Wyo.	16	16	16	15-16	15-16
Reno, Nev.	18	18	18	18	18
Denver, Colo.	17	17	16	16	16
Phoenix, Ariz.	22	21	21	20	20
Bismarck, N. Dak.	17	17	17	17	16
Minneapolis, Minn.	17	17	17	17	16
Omaha, Neb.	19	18	18	17	17
Little Rock, Ark.	17-18	17-18	17	16-17	16-17
Midland, Tex.	18-19	18-19	18	17-18	17-18
New Orleans, La.	18-19	18	17-18	16-18	16-17
Green Bay, Wis.	17	16	16	16	16
Grand Rapids, Mich.	17	17	17	16-17	16
Detroit, Mich.	18	18	18	17	17
Chicago, Ill.	18	18	18	17	17
Nashville, Tenn.	18	17	17	17	16
Burlington, Vt.	16	16	15	15	15
Philadelphia, Penna.	18	17-18	17	17	17
Charleston, W. Va.	18	17	17	17	17
Atlanta, Ga.	18	18	17-18	17	17
Miami, Fla.	21	20	20	19	19
Honolulu, Hawaii	22	21-22	21	20	20
Anchorage, Alas. (1)	18	18	18	18	18

Note: Based upon the site data and study assumptions summarized in Table 10 with summer peaks (except Anchorage), peaking capacity at $100/kw; fan and pump replacement capacity at $150/kw.

(1) Not optimized — winter peak assumed.

TABLE 17

Capital Cost of the Dry Cooling System ($/Kw) for the Economically Optimum Values of Initial Temperature Difference Shown in Table 13 Nuclear-Fueled Generating Unit, Natural-Draft Tower

	$/Kw				
Fixed-Charge Rate:	8%	10%	12%	15%	18%
PLANT SITE					
Seattle, Wash.	24-25	24	24	24	24
San Francisco, Calif.	29	27-28	26	26	26
Los Angeles, Calif.	29	28	28	28	27-28
Great Falls, Mont.	25-26	25	25	25	24
Boise, Ida.	27-28	26	25-26	22-24	22-23
Casper, Wyo.	23-26	23-24	23-24	22-24	22-23
Reno, Nev.	28	27-28	26-27	26	25
Denver, Colo.	27	26-27	26	24-25	24
Phoenix, Ariz.	32	30-31	30	29	28-29
Bismarck, N. Dak.	26-27	25-26	25	25	24-25
Minneapolis, Minn.	28-29	27	26	25	24
Omaha, Neb.	29	28-29	27	25	25
Little Rock, Ark.	26-27	26-27	26-27	25-26	25-26
Midland, Tex.	28-29	28-29	24-28	24-28	23-26
New Orleans, La.	29-31	28-30	28-29	26-29	25-28
Green Bay, Wis.	28	28	24	24	24
Grand Rapids, Mich.	28-29	27-28	26-27	24-25	24
Detroit, Mich.	29-30	28	27	26-27	25
Chicago, Ill.	29-30	28	27	26	25-26
Nashville, Tenn.	27	26-27	26	26	24-26
Burlington, Vt.	27	27	23-27	23	23
Philadelphia, Penna.	29	29	26-28	26	26
Charleston, W. Va.	29-30	29	28-29	27	25-26
Atlanta, Ga.	29	29	28	27	25
Miami, Fla.	35	33	32-33	32	29-30
Honolulu, Hawaii	36-37	35-36	34-35	33-34	33
Anchorage, Alas. (1)	30-31	30	30	30	30

Note: Based upon the site data and study assumptions summarized in Table 10 with summer peaks (except Anchorage), peaking capacity at $100/kw and pump replacement capacity at $225/kw.

(1) Not optimized — winter peak assumed.

TABLE 18

Capital Cost of the Dry Cooling System ($/Kw)
for the Economically Optimum Values
of Initial Temperature Difference Shown in Table 14
Nuclear-Fueled Generating Unit, Mechanical-Draft Tower

	$/Kw				
Fixed-Charge Rate:	8%	10%	12%	15%	18%
PLANT SITE					
Seattle, Wash.	22	22	22	21-22	21
San Francisco, Calif.	24	23	23	23	23
Los Angeles, Calif.	25-26	25	25	24	24
Great Falls, Mont.	23	22-23	22	22	21-22
Boise, Ida.	23	23	23	23	21
Casper, Wyo.	20-22	20-22	20-22	20-21	20
Reno, Nev.	25	25	24	23	22
Denver, Colo.	24	24	23	22	22
Phoenix, Ariz.	30	28	27	25-26	24-25
Bismarck, N. Dak.	24	23	22-23	22	21
Minneapolis, Minn.	24	24	23	22	22
Omaha, Neb.	26	24	24	23	22
Little Rock, Ark.	24-27	23-24	22-24	20-23	20-22
Midland, Tex.	24-26	22-25	22-24	22-24	21-22
New Orleans, La.	26-27	23-27	23-26	22-23	22-23
Green Bay, Wis.	24	22	22	22	22
Grand Rapids, Mich.	24-25	24	23	22	22
Detroit, Mich.	25	25	24	23	23
Chicago, Ill.	26	25	23-24	23	23
Nashville, Tenn.	25	24	24	22-23	22
Burlington, Vt.	24	23	21	21	21
Philadelphia, Penna.	25	24	24	23	23
Charleston, W. Va.	25-27	25	24	24	22
Atlanta, Ga.	26	26	23-25	23-24	23
Miami, Fla.	31	30	30	28	28
Honolulu, Hawaii	33	32	32	31	29-31
Anchorage, Alas. (1)	27	27	27	27	27

Note: Based upon the site data and study assumptions summarized in Table 10 with summer peaks (except Anchorage), peaking capacity at $100/kw; fan and pump replacement capacity at $225/kw.

(1) Not optimized — winter peak assumed.

TABLE 19

Capital Cost of the Dry Cooling System ($/Kw) Plus Capital Cost of Peaking Capacity ($/Kw) for the Economically Optimum Values of Initial Temperature Difference Shown in Table 11 Fossil-Fueled Generating Unit, Natural-Draft Tower

	$/Kw				
Fixed-Charge Rate:	8%	10%	12%	15%	18%
PLANT SITE					
Seattle, Wash.	24	24	24	24	24
San Francisco, Calif.	24	24	24	24	24
Los Angeles, Calif.	25	24-25	24	24	24
Great Falls, Mont.	26	26	26	26	25-26
Boise, Ida.	28	27-28	28	28	28
Casper, Wyo.	26	26	26	26	26
Reno, Nev.	29	29	29	29	29
Denver, Colo.	27	26	26	26	26
Phoenix, Ariz.	32	32	32	31-32	31
Bismarck, N. Dak.	27	27	27	27	27
Minneapolis, Minn.	26	26	26	26	26
Omaha, Neb.	28	28	28	28	28
Little Rock, Ark.	26-27	26	26	26	26
Midland, Tex.	28	28	28	28	28
New Orleans, La.	25-26	25	25	25	25
Green Bay, Wis.	24	24	24	24	24
Grand Rapids, Mich.	25-26	25	25	25	25
Detroit, Mich.	27	27	27	26	26
Chicago, Ill.	26	26	26	26	26
Nashville, Tenn.	26	26	26	26	25-26
Burlington, Vt.	23	23	23	23	23
Philadelphia, Penna.	26	26	26	26	26
Charleston, W. Va.	26	26	25-26	25	25
Atlanta, Ga.	26	26	26	26	26
Miami, Fla.	27	27	26-27	26	26
Honolulu, Hawaii	28	27	27	26	26
Anchorage, Alas. (1)	19	19	19	19	19

Note: Based upon the site data and study assumptions summarized in Table 10 with summer peaks (except Anchorage), peaking capacity at $100/kw and pump replacement capacity at $150/kw.

(1) Not optimized-- winter peak assumed.

TABLE 20

Capital Cost of the Dry Cooling System ($/Kw) Plus
Capital Cost of Peaking Capacity ($/Kw) for the Economically Optimum Values
of Initial Temperature Difference Shown in Table 12
Fossil-Fueled Generating Unit, Mechanical-Draft Tower

	$/Kw				
Fixed-Charge Rate:	8%	10%	12%	15%	18%
PLANT SITE					
Seattle, Wash.	23	23	23	23	23
San Francisco, Calif.	23	23	23	23	23
Los Angeles, Calif.	23	23	23	23	23
Great Falls, Mont.	24	24	24	24	24
Boise, Ida.	26	26	26	26	26
Casper, Wyo.	24	24	24	24	24
Reno, Nev.	27	27	27	27	27
Denver, Colo.	25	24	24	24	24
Phoenix, Ariz.	31	31	31	31	31
Bismarck, N. Dak.	26	26	26	26	26
Minneapolis, Minn.	25	25	25	24-25	24
Omaha, Neb.	27	27	27	27	27
Little Rock, Ark.	26	25-26	25-26	25	25
Midland, Tex.	27	27	27	27	27
New Orleans, La.	24	24	24	24	24
Green Bay, Wis.	23	23	23	23	23
Grand Rapids, Mich.	24	24	24	24	24
Detroit, Mich.	25	25	25	25	25
Chicago, Ill.	25	25	25	25	25
Nashville, Tenn.	25	25	25	25	25
Burlington, Vt.	22	22	22	22	22
Philadelphia, Penna.	25	25	25	25	25
Charleston, W. Va.	25	25	25	25	25
Atlanta, Ga.	25	25	25	25	25
Miami, Fla.	25	25	25	25	25
Honolulu, Hawaii	26	26	25-26	25	25
Anchorage, Alas. (1)	18	18	18	18	18

Note: Based upon the site data and study assumptions summarized in Table 10 with summer peaks (except Anchorage), peaking capacity at $100/kw fan and pump replacement capacity at $150/kw.

(1) Not optimized -- winter peak assumed.

TABLE 21

Capital Cost of the Dry Cooling System ($/Kw) Plus Capital Cost of Peaking Capacity ($/Kw) for the Economically Optimum Values of Initial Temperature Difference Shown in Table 13 Nuclear-Fueled Generating Unit, Natural-Draft Tower

	$/Kw				
Fixed-Charge Rate·	8%	10%	12%	15%	18%
PLANT SITE					
Seattle, Wash.	36	36	36	36	36
San Francisco, Calif.	37-38	37	37	37	37
Los Angeles, Calif.	38	38	38	38	38
Great Falls, Mont.	38-39	38	38	38	38
Boise, Ida.	41-42	41	41	41	41
Casper, Wyo.	38	38	38	38	38
Reno, Nev.	42-43	42	42	42	41-42
Denver, Colo.	40	39-40	39	39	39
Phoenix, Ariz.	47-48	46-47	46	46	45-46
Bismarck, N. Dak.	40	40	40	40	39-40
Minneapolis, Minn.	39-40	39	39	38-39	38
Omaha, Neb.	42	42	42	41	41
Little Rock, Ark.	40-41	40	40	39-40	39
Midland, Tex.	42-43	42	41-42	40-42	40-42
New Orleans, La.	39-40	39-40	39	38-39	38-39
Green Bay, Wis.	37	37	36	36	36
Grand Rapids, Mich.	39	39	39	38	38
Detroit, Mich.	41	40	40	39-40	39
Chicago, Ill.	40-41	40	49	39	39
Nashville, Tenn.	39	39	39	39	39
Burlington, Vt.	36	36	36	35	35
Philadelphia, Penna.	40	40	40	39	39
Charleston, W. Va.	40	40	39	39	38-39
Atlanta, Ga.	40	40	40	40	39
Miami, Fla.	43	42	41	41	41
Honolulu, Hawaii	43	42-43	41-42	41	41
Anchorage, Alas. (1)	30-31	30	30	30	30

Note Based upon the site data and study assumptions summarized in Table 10 with summer peaks (except Anchorage), peaking capacity at $100/kw and pump replacement capacity at $225/kw.

(1) Not optimized -- winter peak assumed.

TABLE 22

Capital Cost of the Dry Cooling System ($/Kw) Plus Capital Cost of Peaking Capacity ($/Kw) for the Economically Optimum Values of Initial Temperature Difference Shown in Table 14
Nuclear-Fueled Generating Unit, Mechanical-Draft Tower

	$/Kw				
Fixed-Charge Rate:	8%	10%	12%	15%	18%
PLANT SITE					
Seattle, Wash.	34	34	34	34	34
San Francisco, Calif.	35	34	34	34	34
Los Angeles, Calif.	35	35	35	35	35
Great Falls, Mont.	36	36	36	36	35
Boise, Ida.	39	38	38	38	38
Casper, Wyo.	36	36	36	36	36
Reno, Nev.	40	39	39	39	39
Denver, Colo.	36	36	36	36	36
Phoenix, Ariz.	45	44	44	43	43
Bismarck, N. Dak.	38	38	38	37	37
Minneapolis, Minn.	37	37	36	36	36
Omaha, Neb.	39	39	39	39	39
Little Rock, Ark.	37-38	37-38	37	37	37
Midland, Tex.	39	39	38-39	38-39	38-39
New Orleans, La.	36	35-36	35-36	35	35
Green Bay, Wis.	35	34	34	34	34
Grand Rapids, Mich.	36	36	36	36	36
Detroit, Mich.	38	38	38	37	37
Chicago, Ill.	38	37	37	37	37
Nashville, Tenn.	37	37	37	36-37	36
Burlington, Vt.	34	33	33	33	33
Philadelphia, Penna.	37	37	37	37	37
Charleston, W. Va.	37	37	36	36	36
Atlanta, Ga.	38	37	37	37	37
Miami, Fla.	39	38	38	38	38
Honolulu, Hawaii	39	39	39	38	38
Anchorage, Alas. (1)	27	27	27	27	27

Note: Based upon the site data and study assumptions summarized in Table 10 with summer peaks (except Anchorage), peaking capacity at $100/kw; fan and pump replacement capacity at $225/kw.

(1) Not optimized -- winter peak assumed.

Tables 23 through 26 show, for optimized installations, the annual costs of the cooling system (including the condensers and all other equipment associated with the cooling system), the capacity necessary to replace loss of turbine capacity at high ambient temperatures, the cooling system auxiliary capacity requirements, and the total plant fuel cost. The above annual costs are presented in mills per kw for the 3 ranges of fuel cost used for each location and for the 5 fixed-charge rates considered for the optimization program. The annual plant costs for other parts of the generating plant, except for the total plant fuel cost which was included above, were not incorporated in the figures listed on Tables 23 through 26. The optimization program evaluated only those parameters affected by the dry-type cooling system. It is possible that the cost of the turbine will be affected to some degree by the varying conditions studied, but it is assumed that the net result will be no increase in cost from a present-day standard design. Some foreign firms claim a reduction in turbine cost due to design for operation at high back pressures.

Tables 27 through 30 show the auxiliary capacity requirements, in mw, for optimized dry-type cooling system installations for the 3 ranges of fuel costs used and for the 5 fixed-charge rates used in the computer analysis program.

TABLE 23

Optimized Total Annual Costs (in Mills per Kwh) Influenced by the Cooling System
800-Mw, Fossil-Fueled Generating Unit
Natural-Draft, Dry-Type Cooling Tower System

	Low Fuel Cost Range					Medium Fuel Cost Range					High Fuel Cost Range				
Fixed-Charge Rate:	8%	10%	12%	15%	18%	8%	10%	12%	15%	18%	8%	10%	12%	15%	18%
PLANT SITE															
Seattle, Wash.	2.64	2.71	2.79	2.90	3.02	3.56	3.63	3.71	3.82	3.93	4.01	4.09	4.17	4.28	4.39
San Francisco, Calif.	2.65	2.73	2.80	2.92	3.03	3.57	3.64	3.72	3.84	3.95	4.03	4.10	4.18	4.30	4.41
Los Angeles, Calif.	2.66	2.74	2.82	2.94	3.05	3.58	3.66	3.74	3.86	3.97	4.04	4.12	4.20	4.32	4.43
Great Falls, Mont.	1.76	1.84	1.92	2.04	2.16	2.67	2.76	2.84	2.96	3.08	3.13	3.21	3.30	3.42	3.54
Boise, Ida.	2.26	2.35	2.44	2.57	2.70	3.18	3.27	3.35	3.49	3.62	3.64	3.73	3.81	3.95	4.08
Casper, Wyo.	1.30	1.38	1.46	1.59	1.71	2.22	2.30	2.39	2.51	2.64	2.68	2.76	2.85	2.97	3.10
Reno, Nev.	2.73	2.82	2.91	3.05	3.19	3.65	3.74	3.83	3.97	4.11	4.11	4.20	4.29	4.43	4.57
Denver Colo.	2.23	2.32	2.40	2.53	2.65	3.15	3.24	3.32	3.45	3.57	3.61	3.70	3.78	3.91	4.03
Phoenix, Ariz.	2.35	2.46	2.56	2.71	2.86	3.28	3.38	3.48	3.64	3.79	3.74	3.84	3.95	4.10	4.25
Bismarck, N. Dak.	1.50	1.58	1.67	1.80	1.92	2.23	2.32	2.40	2.53	2.66	2.69	2.78	2.86	2.99	3.12
Minneapolis, Minn.	2.68	2.76	2.84	2.96	3.09	3.60	3.68	3.76	3.88	4.00	4.06	4.14	4.22	4.34	4.46
Omaha, Neb.	2.72	2.80	2.89	3.03	3.16	3.64	3.73	3.81	3.95	4.06	4.10	4.19	4.27	4.41	4.54
Little Rock, Ark.	2.70	2.79	2.87	2.99	3.12	3.63	3.71	3.80	3.92	4.05	4.09	4.18	4.26	4.39	4.51
Midland, Tex.	2.27	2.36	2.45	2.58	2.71	3.20	3.29	3.38	3.51	3.64	3.66	3.75	3.84	3.98	4.11
New Orleans, La.	2.23	2.31	2.39	2.51	2.63	3.16	3.24	3.32	3.44	3.56	3.62	3.71	3.79	3.91	4.03
Green Bay, Wis.	2.65	2.73	2.80	2.92	3.04	3.57	3.65	3.72	3.84	3.95	4.03	4.10	4.18	4.30	4.41
Grand Rapids, Mich.	2.67	2.76	2.84	2.96	3.08	3.59	3.67	3.76	3.88	4.00	4.05	4.13	4.22	4.34	4.46
Detroit, Mich.	2.69	2.78	2.86	2.99	3.11	3.61	3.70	3.78	3.91	4.03	4.07	4.15	4.24	4.37	4.49
Chicago, Ill	2.69	2.77	2.86	2.98	3.11	3.61	3.69	3.78	3.90	4.03	4.07	4.15	4.24	4.36	4.49
Nashville, Tenn.	2.05	2.13	2.22	2.34	2.46	2.70	2.78	2.86	2.98	3.11	3.16	3.24	3.32	3.44	3.57
Burlington, Vt.	2.64	2.71	2.78	2.90	3.00	3.55	3.63	3.70	3.81	3.92	4.01	4.09	4.16	4.27	4.38
Philadelphia, Penna.	2.69	2.78	2.86	2.98	3.11	3.61	3.70	3.78	3.90	4.03	4.07	4.16	4.24	4.36	4.49
Charleston, W. Va.	1.76	1.85	1.93	2.05	2.17	2.69	2.77	2.85	2.97	3.09	3.15	3.23	3.31	3.43	3.55
Atlanta, Ga.	2.70	2.78	2.87	2.99	3.12	3.62	3.71	3.79	3.91	4.04	4.08	4.17	4.25	4.37	4.50
Miami, Fla.	2.73	2.82	2.90	3.03	3.15	3.66	3.74	3.83	3.96	4.08	4.12	4.21	4.29	4.42	4.55
Honolulu, Hawaii	3.19	3.28	3.36	3.49	3.61	4.12	4.20	4.29	4.42	4.54	4.58	4.67	4.76	4.88	5.01
Anchorage, Alas.	3.04	3.11	3.17	3.26	3.35	3.97	4.03	4.09	4.18	4.27	4.43	4.49	4.55	4.65	4.74

(1) The costs shown in this table reflect the study assumptions as summarized in Table 10.

(2) The costs influenced by the cooling system are: a) the annual capital and operating cost of the cooling system (from the turbine flange outward); b) the annual cost of auxiliary power and energy required for the cooling system; c) the annual cost of replacing capacity and energy lost at high turbine back pressures; and d) the total annual plant fuel cost.

(3) The costs shown in this table do not include the variable and fixed costs, except fuel, related to the basic generating plant (boiler, turbine-generator, auxiliary equipment associated with the boiler and turbine-generator, step-up transformer, switchgear equipment, and associated structures and foundations).

TABLE 24

Optimized Total Annual Costs (in Mills per Kwh) Influenced by the Cooling System
800-Mw, Fossil-Fueled Generating Unit
Mechanical-Draft, Dry-Type Cooling Tower System

Fixed-Charge Rate:	Low Fuel Cost Range					Medium Fuel Cost Range					High Fuel Cost Range				
PLANT SITE	8%	10%	12%	15%	18%	8%	10%	12%	15%	18%	8%	10%	12%	15%	18%
Seattle, Wash.	2.68	2.76	2.84	2.95	3.07	3.61	3.69	3.77	3.88	4.00	4.08	4.16	4.23	4.35	4.47
San Francisco, Calif.	2.69	2.77	2.85	2.97	3.09	3.63	3.70	3.78	3.90	4.02	4.09	4.17	4.25	4.37	4.49
Los Angeles, Calif.	2.71	2.80	2.88	3.00	3.12	3.65	3.73	3.81	3.93	4.05	4.12	4.20	4.28	4.40	4.52
Great Falls, Mont.	1.78	1.86	1.94	2.07	2.19	2.71	2.79	2.87	3.00	3.12	3.17	3.25	3.34	3.46	3.58
Boise, Ida.	2.29	2.38	2.47	2.60	2.74	3.22	3.31	3.40	3.54	3.67	3.69	3.78	3.87	4.00	4.14
Casper, Wyo.	1.31	1.39	1.48	1.60	1.72	2.24	2.33	2.41	2.53	2.66	2.71	2.79	2.88	3.00	3.13
Reno, Nev.	2.76	2.85	2.95	3.09	3.22	3.69	3.79	3.88	4.02	4.15	4.16	4.25	4.34	4.48	4.62
Denver, Colo.	2.26	2.35	2.43	2.56	2.68	3.19	3.28	3.36	3.49	3.61	3.66	3.74	3.83	3.95	4.08
Phoenix, Ariz.	2.40	2.51	2.61	2.77	2.93	3.34	3.45	3.56	3.72	3.87	3.82	3.92	4.03	4.19	4.34
Bismarck, N. Dak.	1.53	1.61	1.70	1.83	1.96	2.27	2.36	2.44	2.58	2.71	2.73	2.82	2.91	3.04	3.17
Minneapolis, Minn.	2.72	2.81	2.89	3.02	3.14	3.65	3.74	3.82	3.95	4.07	4.12	4.20	4.28	4.41	4.54
Omaha, Neb.	2.76	2.86	2.95	3.09	3.22	3.70	3.79	3.88	4.02	4.15	4.16	4.26	4.35	4.48	4.62
Little Rock, Ark.	2.76	2.85	2.93	3.07	3.20	3.70	3.79	3.88	4.01	4.14	4.17	4.26	4.35	4.48	4.62
Midland, Tex.	2.31	2.40	2.49	2.63	2.77	3.25	3.35	3.44	3.58	3.72	3.73	3.82	3.91	4.05	4.19
New Orleans, La.	2.27	2.36	2.44	2.56	2.68	2.22	3.30	3.39	3.51	3.64	3.69	3.78	3.86	3.99	4.11
Green Bay, Wis.	2.69	2.77	2.85	2.97	3.09	3.62	3.70	3.78	3.90	4.02	4.08	4.16	4.24	4.36	4.48
Grand Rapids, Mich.	2.72	2.80	2.89	3.01	3.14	3.65	3.73	3.82	3.94	4.07	4.12	4.20	4.28	4.41	4.53
Detroit, Mich.	2.74	2.82	2.91	3.04	3.17	3.67	3.76	3.84	3.97	4.10	4.13	4.22	4.31	4.44	4.57
Chicago, Ill.	2.73	2.82	2.91	3.03	3.16	3.66	3.75	3.84	3.97	4.09	4.13	4.22	4.30	4.43	4.56
Nashville, Tenn.	2.10	2.18	2.27	2.40	2.53	2.75	2.84	2.92	3.05	3.18	3.22	3.31	3.39	3.52	3.65
Burlington, Vt.	2.68	2.76	2.83	2.95	3.06	3.61	3.69	3.76	3.88	3.99	4.07	4.15	4.23	4.34	4.46
Philadelphia, Penna.	2.74	2.83	2.92	3.05	3.18	3.68	3.76	3.85	3.98	4.11	4.14	4.23	4.32	4.45	4.58
Charleston, W. Va.	1.80	1.89	1.97	2.10	2.22	2.73	2.82	2.90	3.03	3.16	3.20	3.29	3.37	3.50	3.62
Atlanta, Ga.	2.76	2.84	2.93	3.06	3.19	3.69	3.78	3.87	4.00	4.13	4.16	4.25	4.33	4.46	4.59
Miami, Fla.	2.78	2.87	2.96	3.09	3.22	3.73	3.82	3.90	4.04	4.17	4.20	4.29	4.38	4.51	4.64
Honolulu, Hawaii	3.26	3.35	3.43	3.57	3.70	4.20	4.29	4.38	4.51	4.64	4.67	4.76	4.85	4.99	5.12
Anchorage, Alas.	3.07	3.13	3.20	3.29	3.38	4.00	4.06	4.13	4.22	4.31	4.47	4.53	4.59	4.69	4.78

(1) The costs shown in this table reflect the study assumptions as summarized in Table 10.

(2) The costs influenced by the cooling system are: a) the annual capital and operating cost of the cooling system (from the turbine flange outward); b) the annual cost of auxiliary power and energy required for the cooling system; c) the annual cost of replacing capacity and energy lost at high turbine back pressures; and d) the total annual plant fuel cost.

(3) The costs shown in this table do not include the variable and fixed costs, except fuel, related to the basic generating plant (boiler, turbine-generator, auxiliary equipment associated with the boiler and turbine-generator, step-up transformer, switchgear equipment, and associated structures and foundations).

TABLE 25

Optimized Total Annual Costs (in Mills per Kwh) Influenced by the Cooling System
800-Mw, Nuclear-Fueled Generating Unit
Natural-Draft, Dry-Type Cooling Tower System

	Low Fuel Cost Range					Medium Fuel Cost Range					High Fuel Cost Range				
Fixed-Charge Rate:	8%	10%	12%	15%	18%	8%	10%	12%	15%	18%	8%	10%	12%	15%	18%
PLANT SITE															
Seattle, Wash.	1.72	1.83	1.95	2.13	2.30	2.14	2.26	2.38	2.55	2.73	2.68	2.79	2.91	3.09	3.26
San Francisco, Calif.	1.73	1.85	1.97	2.15	2.33	2.16	2.28	2.40	2.58	2.76	2.69	2.81	2.93	3.11	2.29
Los Angeles, Calif.	1.75	1.88	2.00	2.19	2.37	2.18	2.30	2.43	2.61	2.80	2.71	2.84	2.96	3.15	3.33
Great Falls, Mont.	1.77	1.89	2.01	2.20	2.38	2.19	2.32	2.44	2.62	2.81	2.73	2.85	2.97	3.16	3.34
Boise, Ida.	1.83	1.96	2.09	2.29	2.49	2.25	2.39	2.52	2.72	2.91	2.79	2.92	3.05	3.25	3.45
Casper, Wyo.	1.75	1.88	2.00	2.18	2.36	2.19	2.32	2.44	2.62	2.81	2.73	2.86	2.98	3.17	3.35
Reno, Nev.	1.84	1.98	2.11	2.31	2.51	2.27	2.41	2.54	2.74	2.94	2.80	2.94	3.08	3.28	3.48
Denver, Colo.	1.80	1.92	2.05	2.24	2.43	2.22	2.35	2.48	2.67	2.86	2.76	2.89	3.01	3.20	3.39
Phoenix, Ariz.	1.97	2.12	2.27	2.50	2.72	2.41	2.56	2.71	2.93	3.15	2.95	3.10	3.25	3.47	3.70
Bismarck, N. Dak.	1.80	1.93	2.05	2.25	2.44	2.22	2.35	2.48	2.67	2.87	2.76	2.89	3.02	3.21	3.40
Minneapolis, Minn.	1.78	1.91	2.03	2.22	2.41	2.21	2.33	2.46	2.65	2.83	2.74	2.87	2.99	3.15	3.37
Omaha, Neb.	1.83	1.97	2.10	2.30	2.50	2.26	2.40	2.53	2.73	2.93	2.80	2.93	3.07	3.27	3.47
Little Rock, Ark.	1.80	1.93	2.06	2.25	2.45	2.25	2.38	2.51	2.70	2.89	2.79	2.92	3.05	3.25	3.44
Midland, Tex.	1.84	1.97	2.10	2.30	2.50	2.28	2.42	2.56	2.76	2.96	2.83	2.97	3.10	3.31	3.51
New Orleans, La.	1.79	1.92	2.05	2.23	2.42	2.24	2.37	2.50	2.69	2.88	2.78	2.91	3.04	3.23	3.42
Green Bay, Wis.	1.74	1.86	1.98	2.16	2.33	2.17	2.29	2.41	2.58	2.76	2.70	2.82	2.94	3.12	3.30
Grand Rapids, Mich.	1.78	1.90	2.03	2.22	2.40	2.20	2.33	2.46	2.64	2.83	2.74	2.86	2.99	3.18	3.36
Detroit, Mich.	1.80	1.93	2.06	2.26	2.45	2.23	2.36	2.49	2.68	2.87	2.76	2.89	3.02	3.22	3.41
Chicago, Ill.	1.80	1.93	2.06	2.25	2.44	2.22	2.36	2.49	2.68	2.87	2.76	2.89	3.02	3.21	3.40
Nashville, Tenn.	1.80	1.93	2.06	2.25	2.44	2.23	2.36	2.49	2.68	2.87	2.77	2.90	3.03	3.22	3.41
Burlington, Vt.	1.72	1.84	1.95	2.13	2.30	2.15	2.26	2.38	2.56	2.73	2.68	2.80	2.91	3.09	3.26
Philadelphia, Penna.	1.80	1.93	2.06	2.26	2.45	2.23	2.36	2.49	2.68	2.88	2.76	2.90	3.03	3.22	3.41
Charleston, W. Va.	1.79	1.92	2.05	2.24	2.43	2.22	2.35	2.48	2.67	2.86	2.76	2.88	3.01	3.20	3.39
Atlanta, Ga.	1.81	1.94	2.07	2.27	2.46	2.24	2.37	2.50	2.70	2.89	2.78	2.91	3.04	3.23	3.43
Miami, Fla.	1.86	2.00	2.14	2.34	2.54	2.29	2.43	2.57	2.77	2.97	2.83	2.97	3.11	3.31	3.51
Honolulu, Hawaii	1.86	2.00	2.13	2.33	2.53	2.29	2.43	2.56	2.77	2.97	2.83	2.97	3.11	3.31	3.52
Anchorage, Alas.	1.62	1.71	1.81	1.95	2.10	2.05	2.14	2.24	2.38	2.53	2.58	2.68	2.78	2.92	3.07

(1) The costs shown in this table reflect the study assumptions as summarized in Table 10.

(2) The costs influenced by the cooling system are: a) the annual capital and operating cost of the cooling system (from the turbine flange outward); b) the annual cost of auxiliary power and energy required for the cooling system; c) the annual cost of replacing capacity and energy lost at high turbine back pressures; and d) the total annual plant fuel cost.

(3) The costs shown in this table do not include the variable and fixed costs, except fuel, related to the basic generating plant (boiler, turbine-generator, auxiliary equipment associated with the boiler and turbine-generator, step-up transformer, switchgear equipment, and associated structures and foundations).

TABLE 26

Optimized Total Annual Costs (in Mills per Kwh) Influenced by the Cooling System
800-Mw, Nuclear-Fueled Generating Unit
Mechanical-Draft, Dry-Type Cooling Tower System

	Low Fuel Cost Range					Medium Fuel Cost Range					High Fuel Cost Range				
Fixed-Charge Rate:	8%	10%	12%	15%	18%	8%	10%	12%	15%	18%	8%	10%	12%	15%	18%
PLANT SITE															
Seattle, Wash.	1.77	1.89	2.01	2.20	2.38	2.20	2.32	2.45	2.63	2.81	2.74	2.87	2.99	3.17	3.36
San Francisco, Calif.	1.78	1.90	2.03	2.22	2.40	2.22	2.34	2.47	2.65	2.84	2.76	2.89	3.01	3.20	3.38
Los Angeles, Calif.	1.81	1.94	2.07	2.26	2.46	2.25	2.38	2.51	2.70	2.89	2.79	2.92	3.05	3.25	3.44
Great Falls, Mont.	1.81	1.93	2.06	2.26	2.45	2.24	2.37	2.50	2.69	2.88	2.78	2.91	3.04	3.23	3.42
Boise, Ida.	1.87	2.00	2.14	2.35	2.55	2.30	2.44	2.58	2.78	2.99	2.85	2.99	3.12	3.33	3.53
Casper, Wyo.	1.79	1.92	2.04	2.23	2.42	2.24	2.36	2.49	2.68	2.87	2.79	2.91	3.04	3.23	3.43
Reno, Nev.	1.88	2.02	2.16	2.37	2.57	2.31	2.45	2.59	2.80	3.01	2.86	3.00	3.14	3.35	3.55
Denver, Colo.	1.83	1.96	2.10	2.29	2.49	2.27	2.40	2.53	2.73	2.92	2.81	2.94	3.08	3.27	3.47
Phoenix, Ariz.	2.03	2.19	2.35	2.58	2.81	2.47	2.63	2.79	3.02	3.25	3.03	3.19	3.35	3.58	3.81
Bismarck, N. Dak.	1.84	1.97	2.11	2.31	2.51	2.27	2.41	2.54	2.74	2.94	2.82	2.95	3.09	3.29	3.49
Minneapolis, Minn.	1.83	1.96	2.09	2.29	2.48	2.26	2.39	2.53	2.72	2.92	2.80	2.94	3.07	3.27	3.46
Omaha, Neb.	1.88	2.02	2.17	2.37	2.58	2.32	2.47	2.60	2.81	3.02	2.87	3.01	3.15	3.36	3.56
Little Rock, Ark	1.86	2.00	2.14	2.34	2.54	2.32	2.46	2.60	2.80	3.00	2.88	3.01	3.15	3.36	3.56
Midland, Tex.	1.89	2.03	2.16	2.37	2.57	2.34	2.49	2.63	2.84	3.05	2.90	3.05	3.19	3.40	3.61
New Orleans, La.	1.85	1.98	2.11	2.30	2.49	2.30	2.44	2.57	2.77	2.96	2.86	2.99	3.13	3.33	3.53
Green Bay, Wis.	1.79	1.91	2.04	2.22	2.41	2.22	2.35	2.47	2.66	2.84	2.76	2.89	3.01	3.20	3.39
Grand Rapids, Mich.	1.82	1.96	2.09	2.28	2.48	2.26	2.39	2.52	2.72	2.91	2.80	2.94	3.07	3.26	3.46
Detroit, Mich.	1.85	1.98	2.12	2.32	2.52	2.28	2.42	2.56	2.76	2.96	2.83	2.96	3.10	3.30	3.50
Chicago, Ill.	1.84	1.98	2.11	2.31	2.51	2.28	2.42	2.55	2.75	2.95	2.82	2.96	3.09	3.29	3.49
Nashville, Tenn.	1.86	2.00	2.14	2.34	2.54	2.30	2.44	2.57	2.78	2.98	2.85	2.99	3.12	3.33	3.52
Burlington, Vt.	1.77	1.89	2.02	2.20	2.38	2.20	2.33	2.45	2.63	2.81	2.75	2.87	2.99	3.17	3.36
Philadelphia, Penna.	1.86	1.99	2.13	2.33	2.53	2.29	2.43	2.56	2.77	2.97	2.84	2.98	3.11	3.31	3.51
Charleston, W. Va.	1.85	1.98	2.11	2.31	2.51	2.28	2.42	2.55	2.75	2.94	2.83	2.96	3.10	3.29	3.49
Atlanta, Ga.	1.87	2.01	2.15	2.35	2.55	2.31	2.45	2.59	2.79	2.99	2.86	3.00	3.13	3.34	3.54
Miami, Fla.	1.91	2.05	2.19	2.41	2.62	2.35	2.50	2.64	2.85	3.06	2.91	3.05	3.19	3.40	3.61
Honolulu, Hawaii	1.91	2.05	2.20	2.41	2.62	2.35	2.50	2.64	2.85	3.07	2.91	3.06	3.20	3.42	3.63
Anchorage, Alas.	1.63	1.73	1.82	1.97	2.11	2.06	2.16	2.26	2.40	2.55	2.61	2.71	2.80	2.95	3.09

(1) The costs shown in this table reflect the study assumptions as summarized in Table 10.

(2) The costs influenced by the cooling system are: a) the annual capital and operating cost of the cooling system (from the turbine flange outward); b) the annual cost of auxiliary power and energy required for the cooling system; c) the annual cost of replacing capacity and energy lost at high turbine back pressures; and d) the total annual plant fuel cost.

(3) The costs shown in this table do not include the variable and fixed costs, except fuel, related to the basic generating plant (boiler, turbine-generator, auxiliary equipment associated with the boiler and turbine-generator, step-up transformer, switchgear equipment, and associated structures and foundations).

TABLE 27

Auxiliary Capacity Required (in Mw) for Cooling System Pumps
at the Optimum ITD for an 800-Mw, Fossil-Fueled Generating Unit
Natural-Draft, Dry-Type Cooling Tower System

	Low Fuel Cost Range					Medium Fuel Cost Range					High Fuel Cost Range				
Fixed-Charge Rate:	8%	10%	12%	15%	18%	8%	10%	12%	15%	18%	8%	10%	12%	15%	18%
PLANT SITE															
Seattle, Wash.	6.5	6.4	6.4	6.4	6.4	6.5	6.4	6.4	6.4	6.4	6.5	6.5	6.4	6.4	6.4
San Francisco, Calif.	6.5	6.4	6.4	6.4	6.4	6.5	6.5	6.4	6.4	6.4	6.5	6.5	6.5	6.4	6.4
Los Angeles, Calif.	6.8	6.6	6.6	6.6	6.5	6.8	6.8	6.6	6.6	6.6	6.8	6.8	6.6	6.6	6.6
Great Falls, Mont.	6.4	6.4	6.4	6.3	6.1	6.4	6.4	6.4	6.3	6.3	6.5	6.4	6.4	6.3	6.3
Boise, Ida.	6.6	6.5	6.5	6.5	6.4	6.6	6.5	6.5	6.5	6.4	6.6	6.6	6.5	6.5	6.4
Casper, Wyo.	6.3	6.0	6.0	6.0	5.9	6.4	6.4	6.3	6.3	6.0	6.4	6.4	6.4	6.3	6.3
Reno, Nev.	6.5	6.5	6.5	6.4	6.4	6.5	6.5	6.5	6.4	6.4	6.5	6.5	6.4	6.4	6.4
Denver, Colo.	6.6	6.5	6.5	6.4	6.4	6.6	6.5	6.5	6.4	6.4	6.6	6.5	6.5	6.4	6.4
Phoenix, Ariz.	7.7	7.7	7.6	7.2	7.0	7.7	7.7	7.6	7.2	7.2	7.9	7.7	7.6	7.3	7.2
Bismarck, N. Dak.	6.6	6.6	6.5	6.5	6.5	6.6	6.6	6.5	6.5	6.5	6.6	6.6	6.5	6.5	6.5
Minneapolis, Minn.	6.8	6.6	6.6	6.5	6.5	6.8	6.6	6.6	6.5	6.5	6.8	6.6	6.6	6.5	6.5
Omaha, Neb.	6.9	6.8	6.8	6.6	6.6	6.9	6.8	6.8	6.8	6.6	6.9	6.8	6.8	6.8	6.6
Little Rock, Ark.	7.2	7.0	7.0	6.9	6.9	7.3	7.2	7.2	7.0	6.9	7.6	7.3	7.2	7.0	7.0
Midland, Tex.	6.9	6.8	6.8	6.6	6.6	7.0	6.9	6.9	6.8	6.8	7.2	7.0	6.9	6.9	6.8
New Orleans, La.	7.0	6.9	6.8	6.8	6.8	7.2	7.0	7.0	6.9	6.8	7.3	7.2	7.0	7.0	6.9
Green Bay, Wis.	6.5	6.5	6.5	6.4	6.4	6.5	6.5	6.5	6.4	6.4	6.5	6.5	6.5	6.5	6.4
Grand Rapids, Mich.	6.6	6.6	6.6	6.5	6.5	6.8	6.6	6.6	6.5	6.5	6.8	6.6	6.6	6.5	6.5
Detroit, Mich.	6.6	6.6	6.6	6.5	6.5	6.8	6.6	6.6	6.5	6.5	6.8	6.6	6.6	6.5	6.5
Chicago, Ill.	6.6	6.6	6.6	6.5	6.5	6.8	6.6	6.6	6.5	6.5	6.8	6.6	6.6	6.5	6.5
Nashville, Tenn.	7.2	7.0	7.0	6.9	6.8	7.2	7.0	7.0	6.9	6.9	7.2	7.2	7.0	6.9	6.9
Burlington, Vt.	6.6	6.6	6.5	6.5	6.5	6.6	6.6	6.5	6.5	6.5	6.6	6.6	6.5	6.5	6.5
Philadelphia, Penna.	6.9	6.8	6.8	6.6	6.6	6.9	6.8	6.8	6.8	6.6	6.9	6.9	6.8	6.8	6.6
Charleston, W. Va.	6.8	6.8	6.6	6.6	6.5	6.8	6.8	6.6	6.6	6.6	6.8	6.8	6.8	6.6	6.6
Atlanta, Ga.	7.2	7.0	6.9	6.8	6.8	7.2	7.0	6.9	6.9	6.8	7.2	7.0	6.9	6.9	6.8
Miami, Fla.	7.6	7.3	7.3	7.2	7.0	7.7	7.6	7.3	7.2	7.0	7.7	7.6	7.3	7.2	7.0
Honolulu, Hawaii	7.6	7.2	7.2	7.0	6.9	7.6	7.3	7.2	7.0	6.9	7.6	7.3	7.2	7.0	7.0
Anchorage, Alas.	4.7	4.7	4.7	4.7	4.7	4.7	4.7	4.7	4.7	4.7	4.7	4.7	4.7	4.7	4.7

(1) The tabulated values reflect the maximum net power demands required by the pumps to overcome head loss and to circulate sufficient water through the cooling system to remove the heat rejected by the turbine for the range and other operating conditions established at the optimum ITD for the fuel costs and fixed-charge rates used at each location.

(2) Recovery water turbines were assumed to be directly connected to each pump-motor shaft to recover excess pressure head and to control the water pressure in the condenser spray nozzles.

TABLE 28

Auxiliary Capacity Required (in Mw) for Cooling System Pumps and Fans
at the Optimum ITD for an 800-Mw, Fossil-Fueled Generating Unit
Mechanical-Draft, Dry-Type Cooling Tower System

	Low Fuel Cost Range					Medium Fuel Cost Range					High Fuel Cost Range				
Fixed-Charge Rate:	8%	10%	12%	15%	18%	8%	10%	12%	15%	18%	8%	10%	12%	15%	18%
PLANT SITE															
Seattle, Wash.	16.3	15.9	15.6	15.6	15.4	15.9	15.9	15.6	15.6	15.4	15.9	15.9	15.6	15.6	15.6
San Francisco, Calif.	16.3	15.9	15.9	15.6	15.4	16.3	15.9	15.9	15.6	15.6	16.3	15.9	15.9	15.6	15.6
Los Angeles, Calif.	17.2	16.9	16.9	16.9	16.6	17.2	16.9	16.9	16.9	16.6	17.2	16.9	16.9	16.9	16.6
Great Falls, Mont.	16.6	16.6	16.3	15.9	15.9	16.6	16.6	16.3	15.9	15.9	16.9	16.6	16.3	15.9	15.9
Boise, Ida.	17.6	17.2	16.9	16.9	16.9	17.6	17.2	16.9	16.9	16.9	17.6	17.2	16.9	16.9	16.9
Casper, Wyo.	16.3	16.3	15.9	15.9	15.9	16.9	16.6	16.6	16.3	15.9	16.9	16.9	16.6	16.3	16.3
Reno, Nev.	17.2	16.9	16.9	16.6	16.6	17.2	16.9	16.9	16.6	16.6	17.2	16.9	16.9	16.6	16.6
Denver, Colo.	17.6	17.2	16.9	16.9	16.6	17.6	17.2	16.9	16.9	16.6	17.6	17.2	16.9	16.9	16.6
Phoenix, Ariz.	22.2	21.3	20.8	20.4	19.9	22.2	21.3	20.8	20.4	19.9	22.2	21.3	20.8	20.4	19.9
Bismarck, N. Dak.	17.2	16.9	16.9	16.9	16.6	17.2	16.9	16.9	16.9	16.6	17.2	16.9	16.9	16.9	16.6
Minneapolis, Minn.	17.2	16.9	16.9	16.6	16.6	17.2	16.9	16.9	16.9	16.6	17.2	16.9	16.9	16.9	16.6
Omaha, Neb.	18.7	18.3	18.0	17.6	17.2	18.7	18.3	18.0	17.6	17.2	18.7	18.3	18.0	18.0	17.2
Little Rock, Ark.	19.5	19.1	18.7	18.3	18.0	20.4	19.9	19.1	18.7	18.7	20.4	19.9	19.5	19.1	18.7
Midland, Tex.	18.7	18.3	18.0	17.6	17.6	19.5	19.1	18.7	18.4	18.0	19.9	19.5	19.1	18.7	18.3
New Orleans, La.	18.7	18.3	17.6	17.2	16.9	19.5	19.1	18.7	18.3	17.6	19.9	19.5	19.1	18.3	18.0
Green Bay, Wis.	16.9	16.6	16.6	16.3	15.9	16.9	16.6	16.6	16.3	15.9	16.9	16.6	16.3	16.3	15.9
Grand Rapids, Mich.	17.2	16.9	16.9	16.6	16.6	17.2	16.9	16.9	16.9	16.6	17.2	16.9	16.9	16.9	16.6
Detroit, Mich.	17.2	16.9	16.9	16.6	16.6	17.2	16.9	16.9	16.6	16.6	17.2	16.9	16.9	16.6	16.6
Chicago, Ill.	17.2	16.9	16.9	16.6	16.6	17.2	16.9	16.9	16.6	16.6	17.2	16.9	16.9	16.6	16.6
Nashville, Tenn.	20.4	19.5	19.1	18.7	18.3	20.4	19.5	19.1	18.7	18.3	19.9	19.5	19.1	18.7	18.3
Burlington, Vt.	16.9	16.9	16.6	16.6	16.3	16.9	16.9	16.6	16.6	16.3	16.9	16.9	16.6	16.6	16.3
Philadelphia, Penna.	18.3	18.0	17.2	16.9	16.9	18.3	18.0	17.2	16.9	16.9	18.3	17.6	17.2	16.9	16.9
Charleston, W. Va.	18.3	17.6	17.2	16.9	16.9	18.3	17.6	17.2	16.9	16.9	18.3	17.6	17.2	16.9	16.9
Atlanta, Ga.	19.5	18.7	18.7	18.3	18.0	19.5	18.7	18.3	18.0	18.0	19.5	18.7	18.3	18.0	17.6
Miami, Fla.	20.8	20.4	20.4	19.5	19.1	20.8	20.4	19.9	19.5	19.1	20.8	20.4	19.9	19.5	19.1
Honolulu, Hawaii	20.4	19.9	19.1	18.7	19.3	20.4	19.5	19.1	18.7	18.3	20.4	19.9	19.5	18.7	18.3
Anchorage, Alas.	12.0	12.0	12.0	12.0	12.0	12.0	12.0	12.0	12.0	12.0	12.0	12.0	12.0	12.0	12.0

(1) The tabulated values reflect the maximum net power demands required by the fans to pass sufficient air through the heat exchangers and by the pumps to overcome head loss and to circulate sufficient water in the cooling system to remove the heat rejected by the turbine for the range and other operating conditions established at the optimum ITD for the fuel cost and fixed-charge rates used at each location.

(2) Recovery water turbines were assumed to be directly connected to each pump-motor shaft to recover excess pressure head and to control the water pressure in the condenser spray nozzles.

TABLE 29

Auxiliary Capacity Required (in Mw) for Cooling System Pumps
at the Optimum ITD for an 800-Mw, Nuclear-Fueled Generating Unit
Natural-Draft, Dry-Type Cooling Tower System

	Low Fuel Cost Range					Medium Fuel Cost Range					High Fuel Cost Range				
Fixed-Charge Rate:	8%	10%	12%	15%	18%	8%	10%	12%	15%	18%	8%	10%	12%	15%	18%
PLANT SITE															
Seattle, Wash.	8.6	8.6	8.6	8.5	8.5	8.6	8.6	8.6	8.5	8.5	9.5	8.6	8.6	8.6	8.5
San Francisco, Calif.	9.5	8.9	8.6	8.6	8.6	9.5	9.0	8.7	8.6	8.6	9.6	9.5	8.7	8.6	8.6
Los Angeles, Calif.	10.0	9.8	9.6	9.6	9.5	10.0	9.8	9.6	9.6	9.5	10.0	9.8	9.8	9.6	9.6
Great Falls, Mont.	8.6	8.6	8.5	8.5	8.3	8.9	8.6	8.5	8.5	8.5	8.9	8.6	8.5	8.5	8.3
Boise, Ida.	9.0	8.7	8.6	8.6	8.4	9.2	8.7	8.6	8.6	8.3	9.6	8.7	8.7	8.6	8.3
Casper, Wyo.	8.2	8.1	8.0	8.0	8.0	8.5	8.5	8.3	8.1	8.1	8.6	8.5	8.5	8.2	8.1
Reno, Nev.	8.7	8.6	8.5	8.2	8.1	8.9	8.6	8.5	8.2	8.1	8.9	8.7	8.6	8.3	8.2
Denver, Colo.	9.5	9.0	8.9	8.5	8.5	9.6	9.0	8.9	8.6	8.5	9.8	9.2	8.9	8.6	8.5
Phoenix, Ariz.	10.5	10.2	10.2	10.0	9.6	10.5	10.3	10.2	10.0	9.8	10.8	10.3	10.2	10.0	9.8
Bismarck, N. Dak.	9.0	8.7	8.6	8.6	8.3	9.2	8.8	8.7	8.6	8.6	9.2	8.9	8.7	8.6	8.6
Minneapolis, Minn.	9.5	9.2	9.0	8.6	8.5	9.6	9.2	9.0	8.9	8.5	9.8	9.2	9.0	8.9	8.6
Omuha, Neb.	9.8	9.6	9.6	8.7	8.6	9.8	9.8	9.6	8.7	8.7	10.0	9.8	9.6	8.9	8.7
Little Rock, Ark.	10.0	10.0	9.8	9.6	9.6	10.2	10.2	10.0	9.8	9.8	10.5	10.2	10.2	10.0	9.8
Midland, Tex.	9.6	9.6	8.7	8.6	8.5	10.0	9.8	9.6	9.6	8.7	10.2	10.0	9.8	9.6	9.6
New Orleans, La.	10.2	10.0	10.0	9.2	9.0	10.5	10.3	10.3	10.0	10.0	10.8	10.5	10.3	10.2	10.0
Green Bay, Wis.	9.6	9.5	8.6	8.6	8.5	9.6	9.6	8.6	8.6	8.5	9.6	9.6	8.6	8.6	8.5
Grand Rapids, Mich.	9.6	9.2	9.0	8.6	8.5	9.6	9.2	9.0	8.6	8.5	9.8	9.5	9.2	8.9	8.6
Detroit, Mich.	9.5	9.2	9.0	8.6	8.5	9.6	9.2	9.0	8.9	8.5	9.8	9.2	9.0	8.9	8.5
Chicago, Ill.	9.6	9.2	9.0	8.6	8.5	9.6	9.5	9.0	8.6	8.5	9.8	9.5	9.2	8.6	8.5
Nashville, Tenn.	10.2	10.0	10.0	9.8	9.8	10.2	10.2	10.0	9.8	9.8	10.2	10.2	10.0	10.0	9.8
Burlington, Vt.	9.6	9.6	9.6	8.6	8.6	9.6	9.6	9.6	8.6	8.6	9.6	9.6	9.6	8.6	8.6
Philadelphia, Penna.	10.0	9.8	9.6	9.0	8.9	10.0	10.0	9.8	9.2	8.9	10.0	10.0	9.8	9.2	9.0
Charleston, W. Va.	10.0	9.8	9.6	9.2	8.9	10.0	9.8	9.6	9.2	9.0	10.2	10.0	9.6	9.2	9.0
Atlanta, Ga.	10.2	10.2	10.0	9.8	9.6	10.2	10.2	10.0	9.8	9.6	10.3	10.2	10.0	10.0	9.8
Miami, Fla.	11.4	11.0	10.5	10.5	10.3	11.4	11.0	10.8	10.5	10.3	11.4	11.0	10.8	10.5	10.3
Honolulu, Hawaii	11.0	10.8	10.5	10.3	10.2	11.0	10.8	10.5	10.3	10.2	11.2	11.0	10.8	10.5	10.3
Anchorage, Alas.	7.0	7.0	7.0	7.0	7.0	7.0	7.0	7.0	7.0	7.0	7.0	7.0	7.0	7.0	7.0

(1) The tabulated values reflect the maximum net power demands required by the pumps to overcome head loss and to circulate sufficient water through the cooling system to remove the heat rejected by the turbine for the range and other operating conditions established at the optimum ITD for the fuel costs and fixed-charge rates used at each location.

(2) Recovery water turbines were assumed to be directly connected to each pump-motor shaft to recover excess pressure head and to control the water pressure in the condenser spray nozzles

TABLE 30

Auxiliary Capacity Required (in Mw) for Cooling System Pumps and Fans
at the Optimum ITD for an 800-Mw, Nuclear-Fueled Generating Unit
Mechanical-Draft, Dry-Type Cooling Tower System

	Low Fuel Cost Range					Medium Fuel Cost Range					High Fuel Cost Range				
Fixed-Charge Rate:	8%	10%	12%	15%	18%	8%	10%	12%	15%	18%	8%	10%	12%	15%	18%
PLANT SITE															
Seattle, Wash.	23.5	23.1	23.1	22.7	22.7	23.5	23.1	23.1	22.7	22.7	23.5	23.1	23.1	22.7	22.7
San Francisco, Calif.	23.9	23.5	23.5	23.1	23.1	23.9	23.5	23.5	23.1	23.1	23.9	23.5	23.5	23.1	23.1
Los Angeles, Calif.	26.8	25.8	25.8	25.3	25.3	26.3	25.8	25.8	25.3	24.8	26.3	25.8	25.8	25.3	24.8
Great Falls, Mont.	23.5	23.1	23.1	22.7	21.9	23.5	23.5	23.1	22.7	21.9	23.5	23.5	23.1	22.7	21.9
Boise, Ida.	25.3	23.9	23.9	23.5	22.7	25.3	23.9	23.9	23.5	22.7	25.3	23.9	23.9	23.5	22.7
Casper, Wyo.	21.9	21.9	21.9	21.6	21.6	23.5	23.1	22.3	21.9	21.9	23.9	23.1	23.1	22.3	21.9
Reno, Nev.	24.4	23.9	23.1	22.3	21.9	24.4	23.9	23.5	22.3	21.9	24.4	23.9	23.5	22.3	21.9
Denver, Colo.	25.3	25.3	24.8	23.5	23.1	25.8	25.3	24.8	23.5	23.1	25.8	25.3	24.8	23.5	23.1
Phoenix, Ariz.	30.9	29.1	28.5	25.8	25.3	30.9	29.1	28.5	26.3	25.3	30.9	29.1	28.5	26.8	25.8
Bismarck, N. Dak.	24.8	23.9	23.5	22.7	22.3	24.8	23.9	23.5	22.7	22.3	24.8	23.9	23.9	22.7	22.3
Minneapolis, Minn.	25.3	25.3	24.4	23.1	23.1	25.3	25.3	24.4	23.1	23.1	25.3	25.3	24.4	23.1	23.1
Omaha, Neb.	26.8	25.3	24.8	23.9	23.5	26.9	25.3	24.8	23.9	23.5	26.8	25.3	24.8	23.9	23.5
Little Rock, Ark.	27.4	26.3	25.8	23.9	23.9	28.5	27.9	26.8	26.3	25.3	30.9	27.9	27.9	26.8	25.8
Midland, Tex.	25.8	24.4	23.9	23.5	23.1	27.9	26.8	25.8	24.4	23.9	27.9	27.4	26.3	25.8	24.4
New Orleans, La.	28.5	25.8	25.3	24.8	24.4	29.7	29.1	28.5	25.8	25.3	30.3	29.7	29.1	25.8	25.8
Green Bay, Wis.	25.3	23.5	23.5	23.1	23.1	25.3	23.5	23.5	23.1	23.1	25.3	23.5	23.5	23.1	23.1
Grand Rapids, Mich.	25.3	25.3	24.4	23.1	23.1	25.8	25.3	24.4	23.1	23.1	25.8	25.3	24.4	23.1	23.1
Detroit, Mich.	25.3	24.8	24.4	23.1	22.7	25.3	24.8	24.4	23.1	23.1	25.3	25.3	24.4	23.1	23.1
Chicago, Ill.	25.8	25.3	23.5	23.1	22.7	25.8	25.3	23.5	23.1	23.1	25.8	25.3	23.5	23.1	23.1
Nashville, Tenn.	28.5	27.9	27.4	25.8	25.3	28.5	27.9	27.4	25.8	25.3	28.5	27.9	27.4	26.3	25.3
Burlington, Vt.	26.3	25.3	23.5	23.5	23.1	26.3	25.3	23.5	23.5	23.1	26.3	25.3	23.5	23.5	23.1
Philadelphia, Penna.	25.8	25.3	25.3	24.4	23.9	25.8	25.3	25.3	24.4	23.9	25.8	25.3	25.3	24.4	23.9
Charleston, W. Va.	27.9	25.8	25.3	24.8	23.5	27.9	25.8	25.3	24.8	23.5	26.3	25.8	25.3	24.8	23.5
Atlanta, Ga.	28.5	27.9	25.8	25.3	25.3	28.5	27.9	26.3	25.3	25.3	28.5	27.9	27.4	25.8	25.3
Miami, Fla.	33.0	31.6	30.9	29.7	29.7	33.0	31.6	30.9	29.7	29.7	33.0	31.6	30.9	29.7	29.7
Honolulu, Hawaii	31.6	30.9	30.3	29.7	27.9	31.6	30.9	30.3	29.7	27.9	31.6	30.9	30.9	29.7	29.1
Anchorage, Alas.	18.3	18.3	18.3	18.3	18.3	18.3	18.3	18.3	18.3	18.3	18.3	18.3	18.3	18.3	18.3

(1) The tabulated values reflect the maximum net power demands required by the fans to pass sufficient air through the heat exchangers and by the pumps to overcome head loss and to circulate sufficient water in the cooling system to remove the heat rejected by the turbine for the range and other operating conditions established at the optimum ITD for the fuel cost and fixed-charge rates used at each location.

(2) Recovery water turbines were assumed to be directly connected to each pump-motor shaft to recover excess pressure head and to control the water pressure in the condenser spray nozzles.

SECTION XI

DISCUSSION OF RESULTS

General

The results of the economic optimization of dry-type cooling systems have been presented in Section X of this report, and those results reflect the effects of the basic study assumptions and the method of analysis.

As shown on Figures 42 and 43, the economically optimum ITD values, in °F, for fossil-fueled generating units were found to range from the mid 50's to the low 60's in the cooler portions of the United States. In the warmer areas of the country, these optimum ITD values were found to lie in the mid 40's to mid 50's. For nuclear-fueled plants, Figures 44 and 45 show the optimum ITD values to range from the high 50's to mid 60's in the cooler areas and from the high 40's to the mid 60's in the warmer areas.

The optimum ITD values for fossil-fueled plants shown on Figures 42 and 43 may be compared to the design ITD values which are summarized in Table 31 for one United States generating plant and four European plants which are now in operation.

The economic optimization analyses indicate that in the United States the optimum result would be obtained by sizing the dry cooling system so that some loss of capacity would be experienced in hot weather. The analyses indicate that it would be more economical to replace the lost capacity from other generating sources than to increase the size of the cooling system to reduce the capacity losses. As shown on Figures 46 and 47, the capacity losses at the optimum ITD for fossil-fueled units were generally on the order of 5 to 10 percent of rated capacity and the maximum value found for the sites studied was 12.6 percent. The capacity losses at the optimum ITD were found to be somewhat higher for nuclear-fueled units ranging up near 15 percent in many cases. The maximum capacity loss at an optimum ITD was found to be 19.9 percent for the sites studied.

The capital costs of the dry cooling system at optimum ITD as summarized on Figures 50 and 51 for fossil-fueled units were found to range from slightly below $14 per kw to about $25 per kw for mechanical-draft systems and were found to be slightly higher for natural-draft systems, $15 to $27 per kw. The capital cost of dry cooling systems at optimum ITD for nuclear-fueled units is summarized on Figures 52 and 53. These nuclear plant values are about 50 percent higher per kw than the figures for the fossil-fueled plants reflecting the greater heat rejection of the nuclear units. The dry cooling system costs include all costs of the generating

TABLE 31

Initial Temperature Differences of Dry Cooling Systems
Existing Installations Visited

Power Plant			Cooling System	
	Dry-Cooled Generating Units			
Name	No.	Capacity per Unit	Type	Design ITD (°F)
Rugeley (England)	1	120 mw	Natural draft Indirect cooling system	35 (1)
Ibbenbüren (Germany)	1	150 mw	Natural draft Indirect cooling system	50
Volkswagen (Germany)	3	50 mw	Mechanical draft Direct condensing system	51
Gyōngyös (Hungary)	2 2	100 mw 200 mw	Natural draft Indirect cooling system	46 47
Neil Simpson (Wyoming, U.S.)	1	20 mw	Mechanical draft Direct condensing system	55

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(1) Not optimized.

plant from the turbine flange outward, and therefore include condenser costs, the costs of cooling system pumps, piping and valves, and cooling tower costs.

Although the capital cost of the mechanical-draft cooling systems was found to be slightly lower than the capital cost of the natural-draft cooling systems, the economic analyses indicated that the annual cost of the natural-draft systems would be slightly lower than the annual cost of the mechanical-draft systems due to the power and energy requirements of the mechanical-draft fans. This cost difference in favor of the natural-draft systems was very small, less than 0.1 mills per kwh. A detailed study for a particular site and a particular set of conditions would be required in order to select the type of dry cooling system to be used at that site. In some cases, particularly if there is a shortage of capital, a mechanical-draft system may be selected over a natural-draft system on the basis of the capital cost difference.

The combined capital cost of the dry cooling system and the required peaking capacity, both at optimum ITD, as shown on Figures 54 and 55 for fossil-fueled units is generally in the range of $22 to $28 per kw. The corresponding values for nuclear-fueled units are shown on Figures 56 and 57.

The effects of the various parameters on the economic optimization analyses have been studied and are discussed below.

Effect of Fixed-Charge Rate

The effect of increasing the fixed-charge rate is to increase the value of the economically optimum ITD and, therefore, to reduce the cooling system investment. This effect is clearly shown in Tables 11, 12, 13 and 14. In these tables, the optimum ITD at a fixed-charge rate of 18 percent is a few degrees higher than the optimum ITD at a fixed-charge rate of 8 percent.

Effect of Fuel Cost

The effect of increasing the plant fuel cost is to reduce the optimum ITD. This reduction in the optimum ITD increases the cooling system investment and improves the plant efficiency. The range of fuel costs investigated in our analyses had only a minor effect on the optimum ITD. In many cases, as shown in Tables 11 through 14, the optimum ITD was not sufficiently affected by the fuel cost to change the ITD by a full degree F. In other cases, the fuel cost difference caused the optimum ITD value to vary over a range of $1^{\circ}F$ to $4^{\circ}F$ for a given fixed-charge rate.

In order to test the effect of varying the fuel cost over a somewhat wider range, some supplemental runs were made for the Chicago site assuming fossil-fuel costs ranging from 10¢ to 80¢ per million Btu. The results of these analyses are summarized in Table 32.

TABLE 32

Effect of Fuel Cost on Optimum ITD
(Chicago, fossil-fueled plant, 15% fixed-charge rate)

Fuel Cost (¢/10^6 Btu)	Optimum ITD (°F) Natural Draft	Optimum ITD (°F) Mechanical Draft
10	57	61
25	57	61
35	57	61
40	57	61
80	55	59

Effect of Air Temperatures

The results of the economic optimization analyses indicated that the optimum ITD was affected both by the general level of ambient air temperatures and by the summer high temperatures. The results have been analyzed by plotting optimum ITD values as a function of both the median air temperature and the air temperature which is equalled or exceeded only 10 hours each year. Figure 58 shows such a plot for fossil-fueled units with natural-draft towers assuming a fixed-charge rate of 15 percent and a fuel cost of 25¢ per million Btu. Figures 59 through 61 show this type of plot for mechanical-draft fossil, natural-draft nuclear, and mechanical-draft nuclear generating units. The grouping of values on these figures indicates the general relationship between these parameters, but there is some scatter of the data. Apparently, this scatter is due primarily to the differences in site elevations and in capital cost multipliers which were reflected in the results of the basic analyses. Therefore, supplemental analyses were made using the actual air temperature data from all the sites and assuming uniform elevations, capital cost multipliers and fuel costs. The results of these analyses are shown in Figures 62 through 65. Isolines of optimum ITD values have been drawn on these figures to indicate the nature of the relationship between the optimum ITD and air temperatures based on the results of the studies reported herein. The isolines were drawn as straight lines which apparently fit the data, but it is recognized that additional sites should be investigated to better define this relationship between the optimum ITD and ambient air temperatures.

Effect of Assumptions as to Lost Capacity

As discussed in Section IX, the assumptions as to the replacement of lost generating capacity at high ambient air temperatures and the cost of that capacity have a significant effect on the economic optimization. The results reported herein are based on the assumption that capacity lost at high air temperatures must be

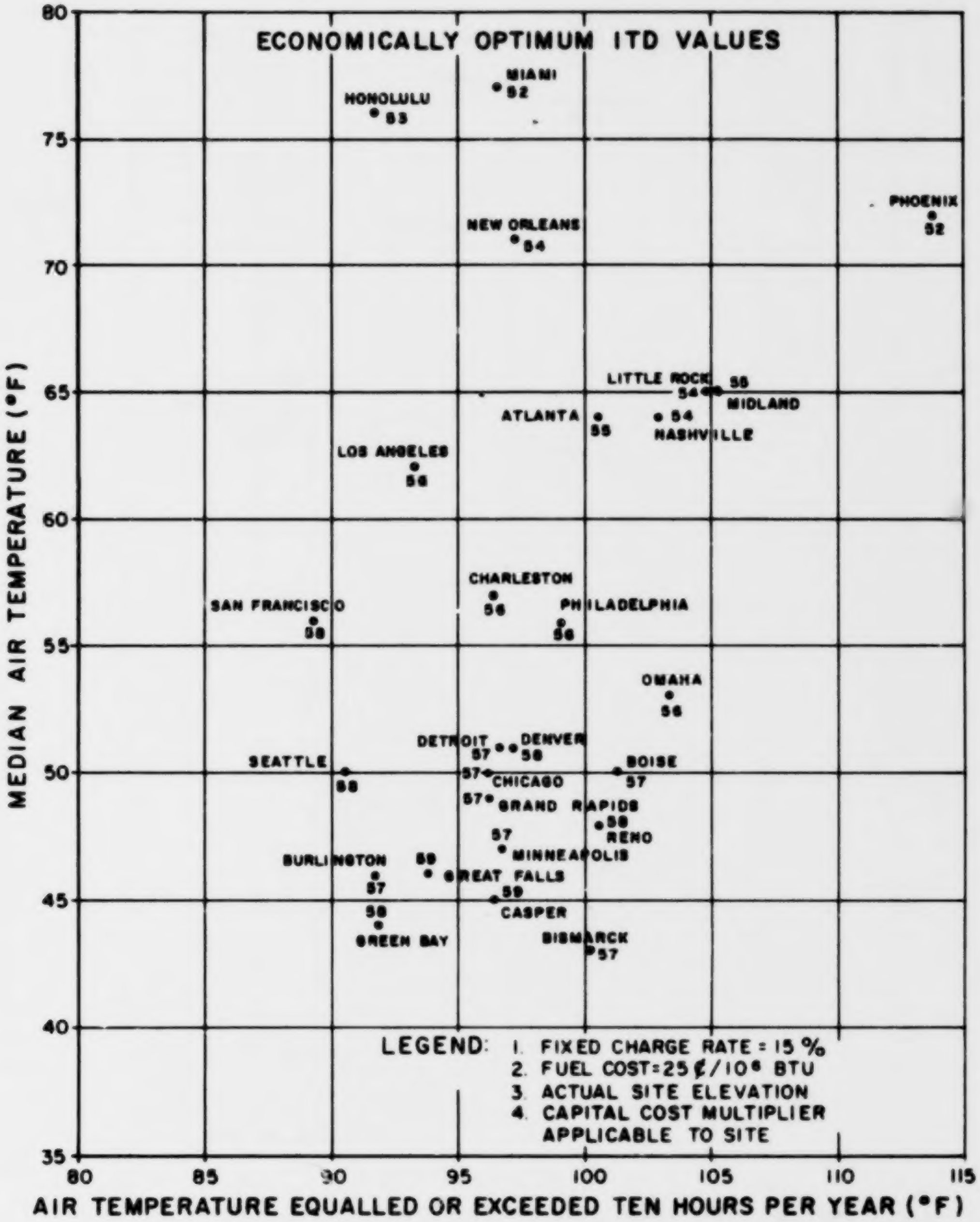

FIGURE 58—RELATIONSHIP OF ECONOMICALLY OPTIMUM INITIAL TEMPERATURE DIFFERENCE TO AMBIENT AIR TEMPERATURES FOR THE SITES STUDIED —— NATURAL-DRAFT DRY COOLING SYSTEM FOR A FOSSIL-FUELED 800 MW GENERATING UNIT

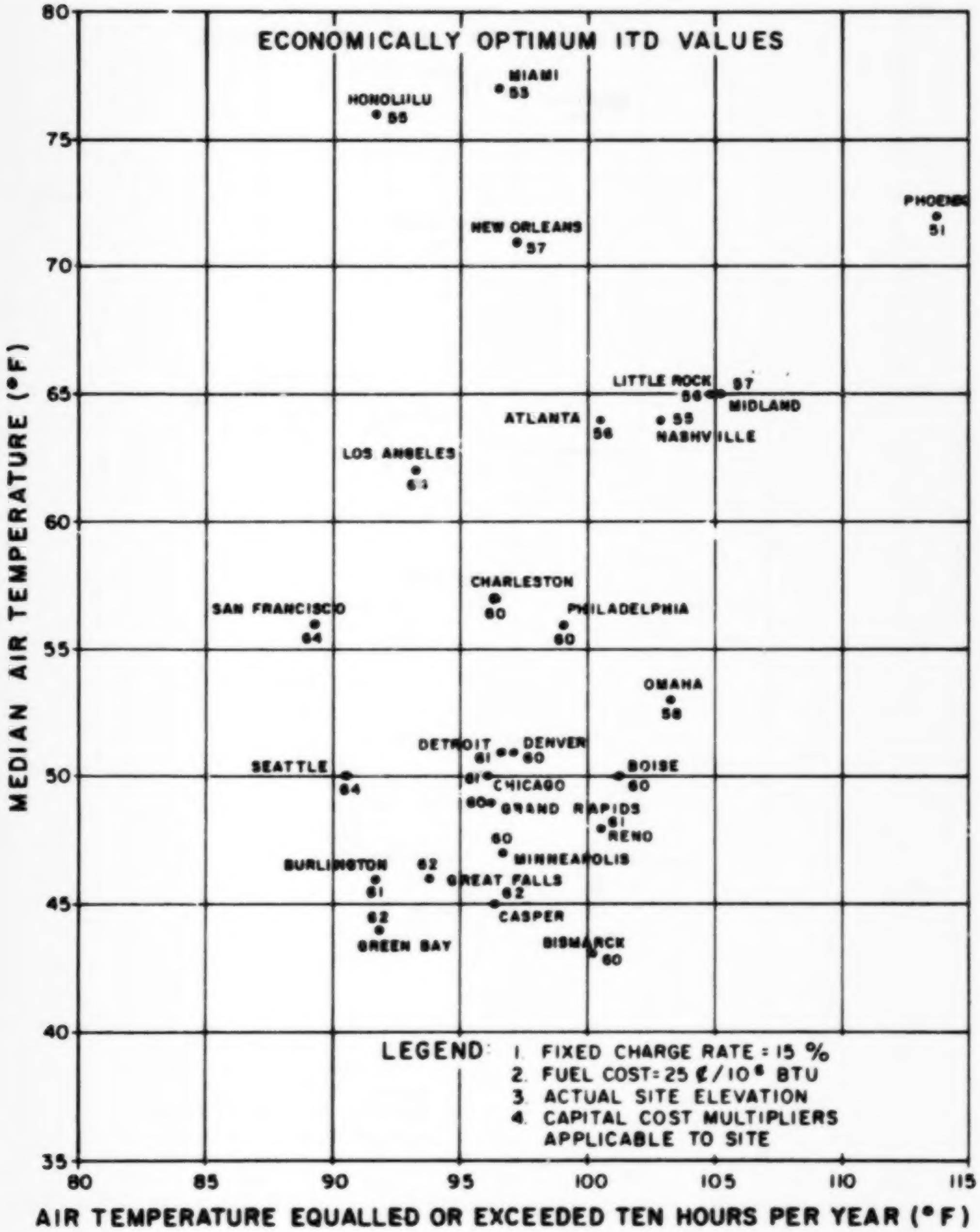

FIGURE 59—RELATIONSHIP OF ECONOMICALLY OPTIMUM INITIAL TEMPERATURE DIFFERENCE TO AMBIENT AIR TEMPERATURES FOR THE SITES STUDIED—MECHANICAL-DRAFT DRY COOLING SYSTEM FOR A FOSSIL-FUELED 800 MW GENERATING UNIT

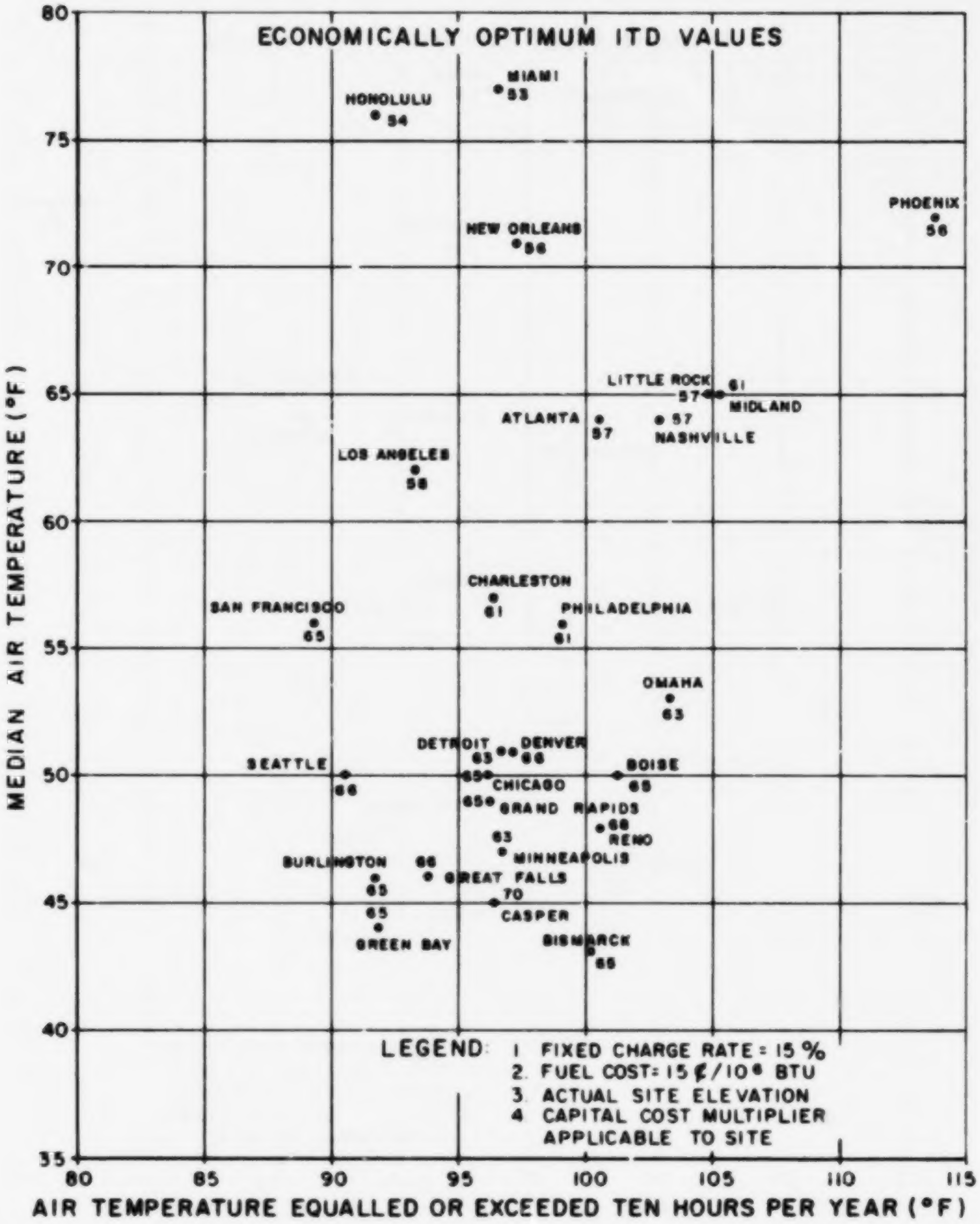

FIGURE 60—RELATIONSHIP OF ECONOMICALLY OPTIMUM INITIAL TEMPERATURE DIFFERENCE TO AMBIENT AIR TEMPERATURES FOR THE SITES STUDIED — NATURAL-DRAFT DRY COOLING SYSTEM FOR A NUCLEAR-FUELED 800 MW GENERATING UNIT

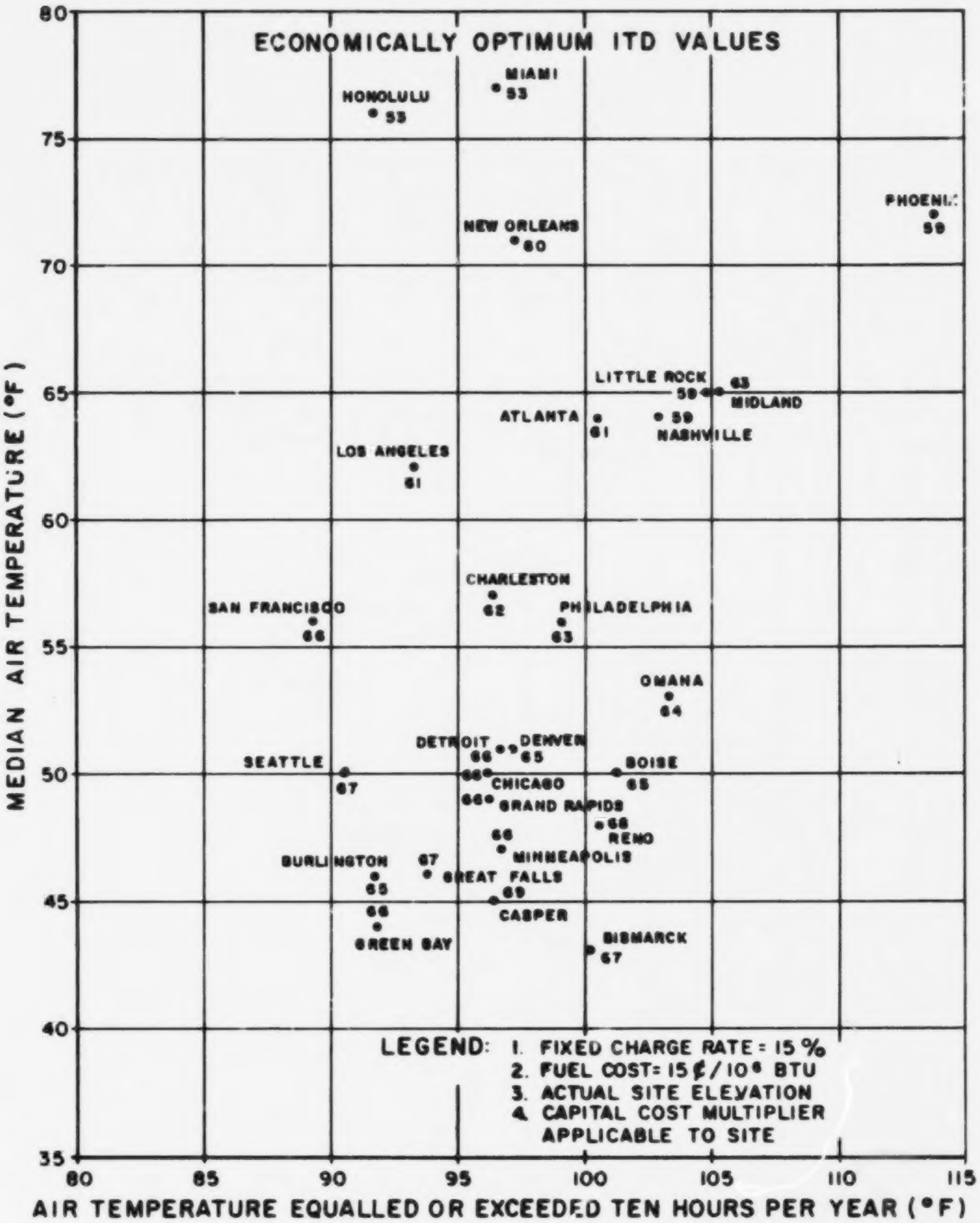

FIGURE 61—RELATIONSHIP OF ECONOMICALLY OPTIMUM INITIAL TEMPERATURE DIFFERENCE TO AMBIENT AIR TEMPERATURES FOR THE SITES STUDIED—MECHANICAL-DRAFT DRY COOLING SYSTEM FOR A NUCLEAR-FUELED 800 MW GENERATING UNIT

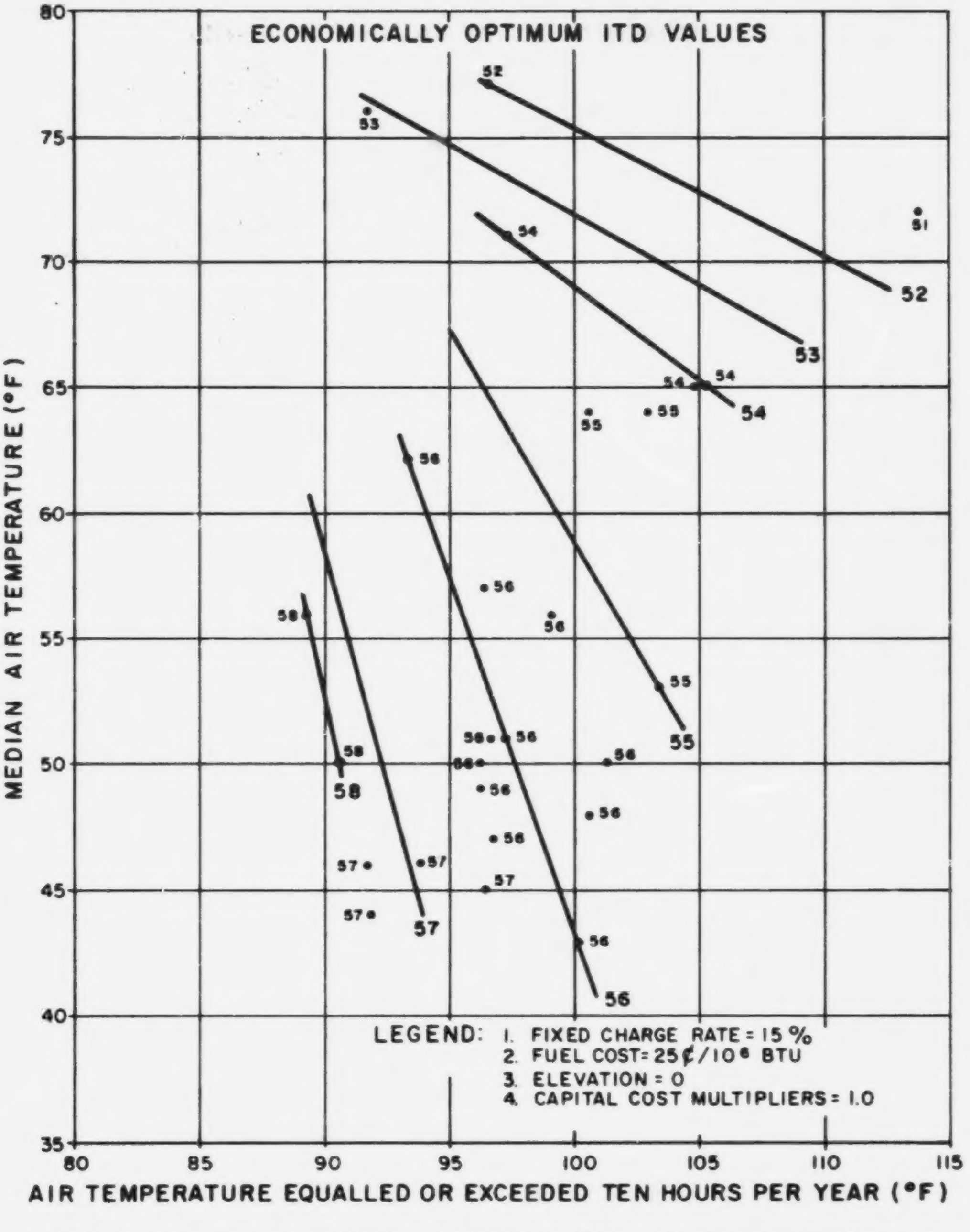

FIGURE 62—RELATIONSHIP OF ECONOMICALLY OPTIMUM INITIAL TEMPERATURE DIFFERENCE TO AMBIENT AIR TEMPERATURES AT SEA-LEVEL ELEVATION—NATURAL-DRAFT DRY COOLING SYSTEM FOR A FOSSIL-FUELED 800 MW GENERATING UNIT

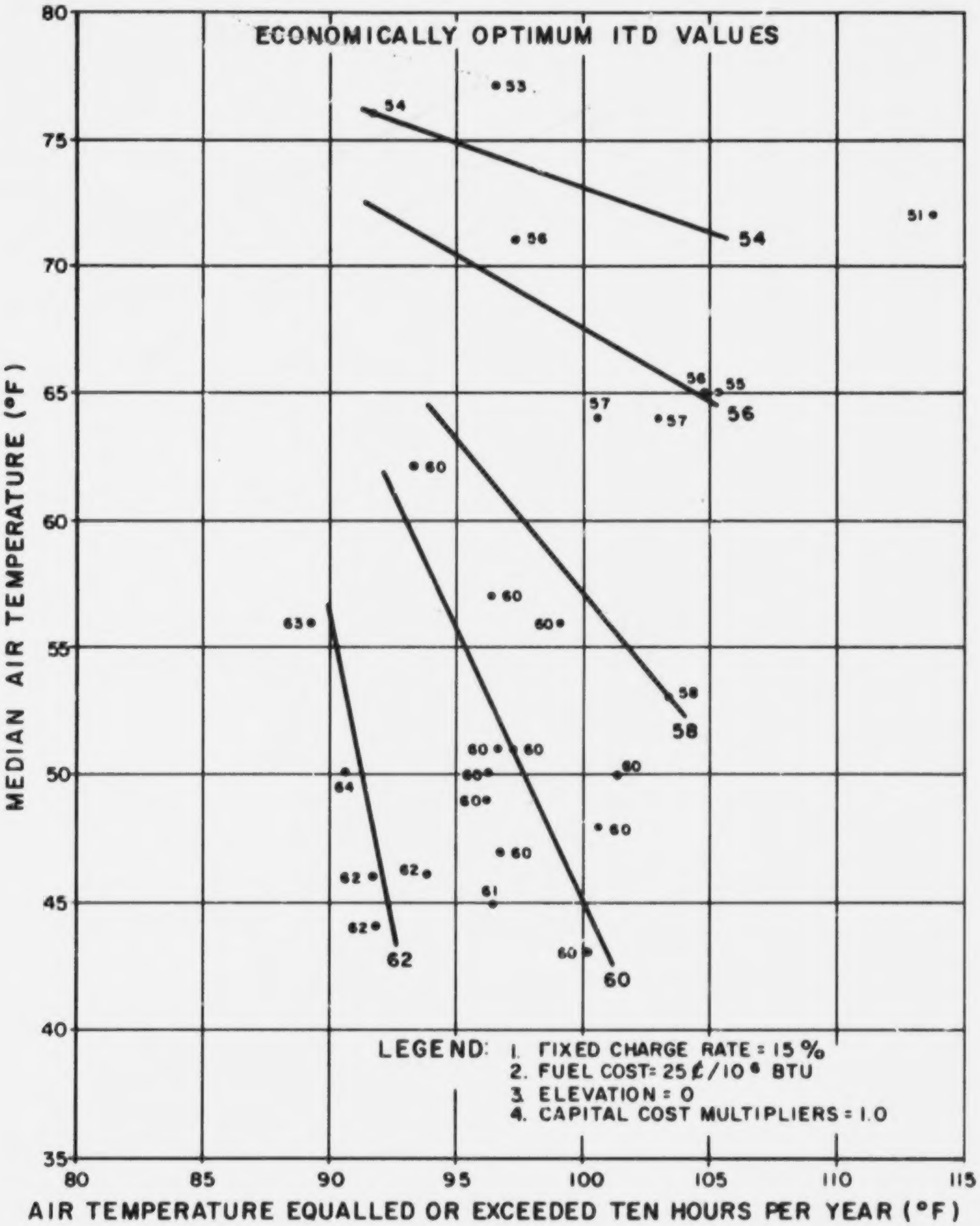

FIGURE 63—RELATIONSHIP OF ECONOMICALLY OPTIMUM INITIAL TEMPERATURE DIFFERENCE TO AMBIENT AIR TEMPERATURES AT SEA-LEVEL ELEVATION— MECHANICAL- DRAFT DRY COOLING SYSTEM FOR A FOSSIL-FUELED 800 MW GENERATING UNIT

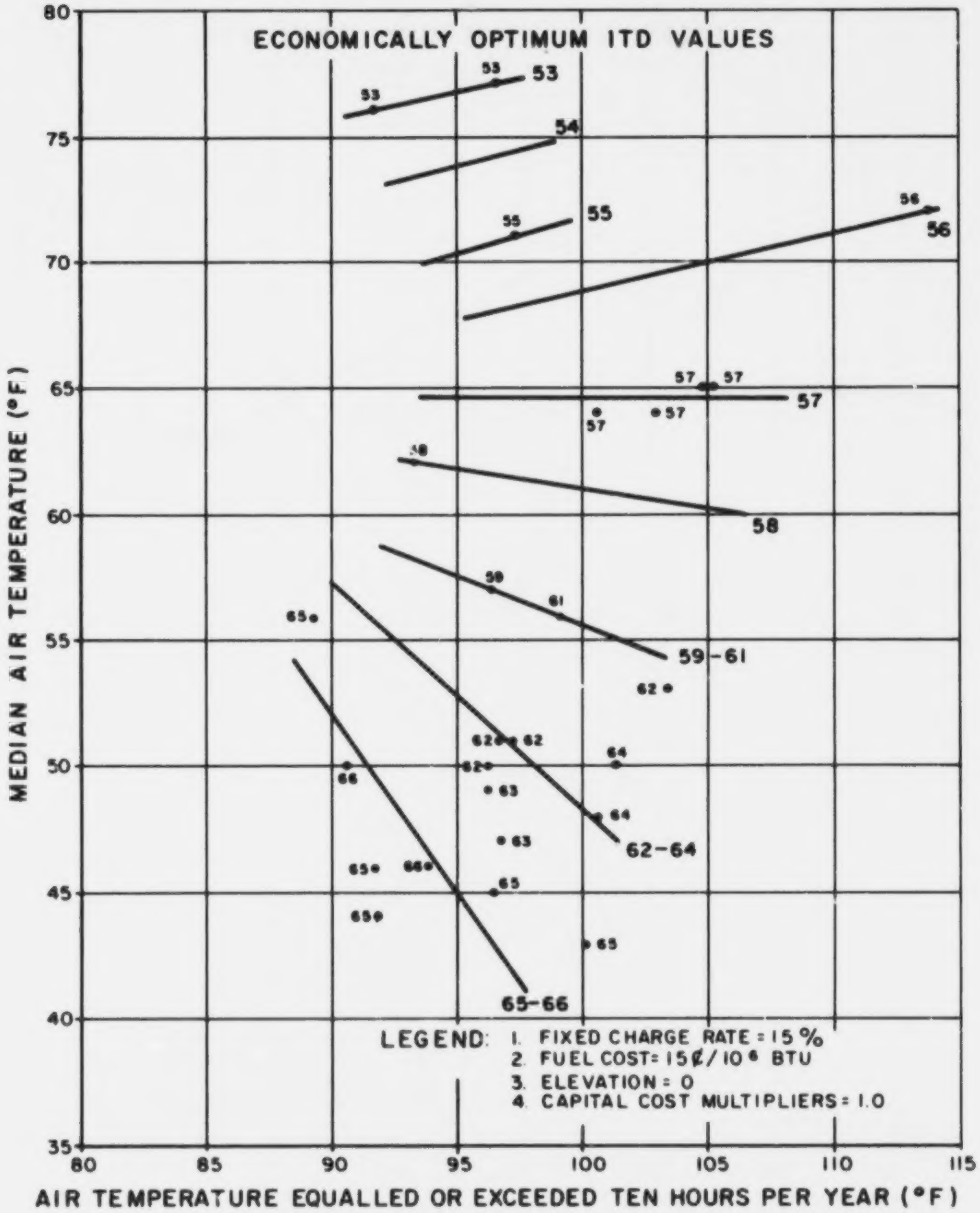

FIGURE 64—RELATIONSHIP OF ECONOMICALLY OPTIMUM INITIAL TEMPERATURE DIFFERENCE TO AMBIENT AIR TEMPERATURES AT SEA-LEVEL ELEVATION— NATURAL- DRAFT DRY COOLING SYSTEM FOR A NUCLEAR-FUELED 800 MW GENERATING UNIT

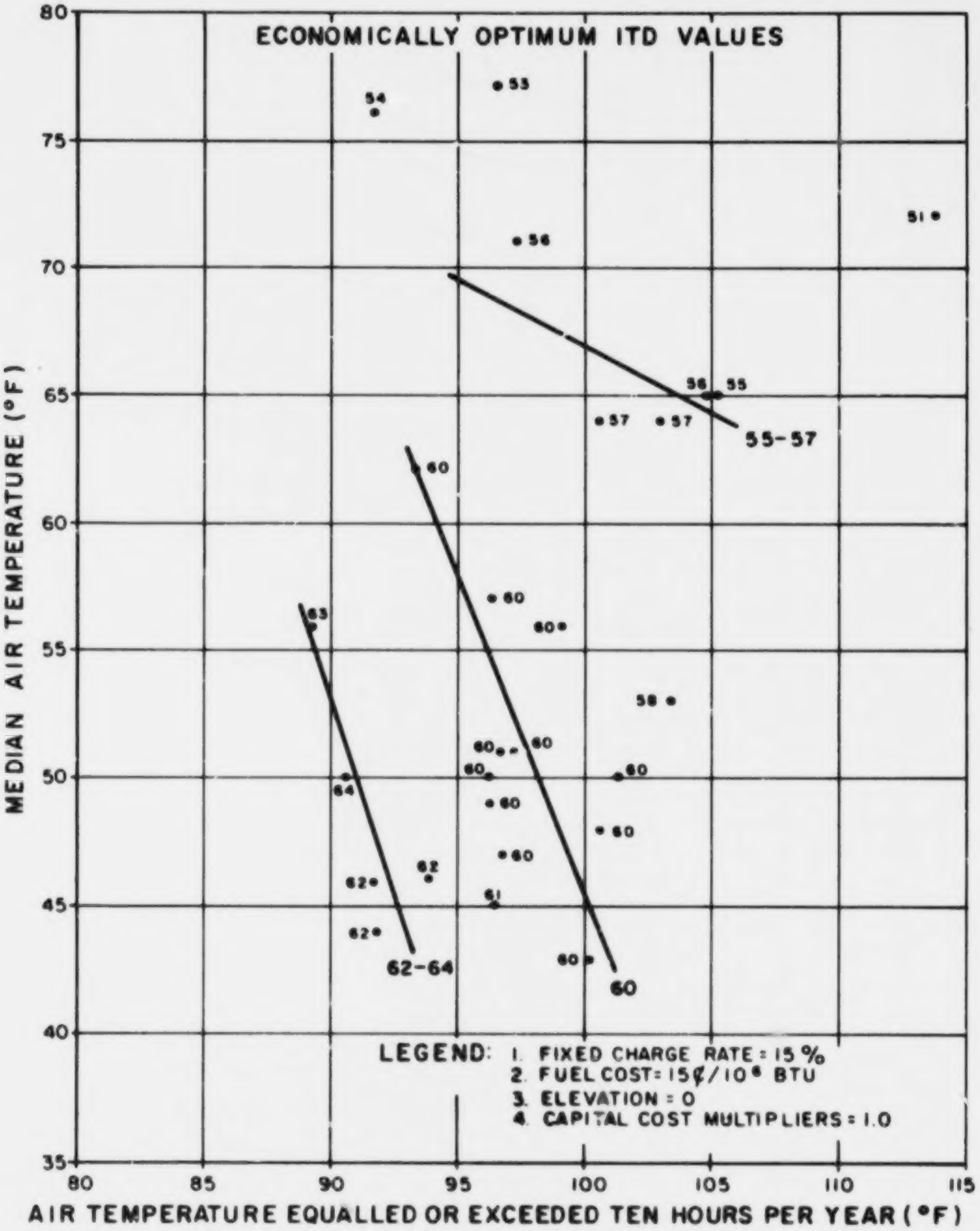

FIGURE 65—RELATIONSHIP OF ECONOMICALLY OPTIMUM INITIAL TEMPERATURE DIFFERENCE TO AMBIENT AIR TEMPERATURES AT SEA-LEVEL ELEVATION — MECHANICAL-DRAFT DRY COOLING SYSTEM FOR A NUCLEAR-FUELED 800 MW GENERATING UNIT

replaced at all sites other than the Anchorage, Alaska site and that the cost of that replacement capacity is $100 per kw. The effect of varying the cost assumption is indicated by the results of supplemental analyses made for the Chicago site. Assuming a base condition for a fossil-fueled plant using a fuel cost of 35¢ per million Btu and a fixed-charge rate of 15 percent, the effect of three different peaking capacity costs—$75 per kw, $100 per kw and $150 per kw—was studied. The results of these analyses are indicated in Table 33.

TABLE 33

Effect of Peaking Capacity Cost on Optimum ITD
(fossil-fueled plant, Chicago,
fuel cost - 35¢ per million Btu, fixed-charge rate - 15%)

Peaking Capacity Cost ($/kw)	Optimum ITD (°F)	
	Natural Draft	Mechanical Draft
75	58	64
100	57	61
150	54	55

Any method of reducing the capital cost of replacing the lost capacity such as the use of dual inlet turbines, or the bypassing of feedwater heaters would tend to increase the optimum ITD and, therefore, reduce the cooling system investment. It is recognized that it may be possible to replace the lost capacity for considerably less than the lowest cost shown in Table 33, $75 per kw.

The effect of varying the assumption as to winter or summer peak was also investigated. If a winter peak is assumed for Chicago, and therefore the lost capacity is not replaced, the optimum ITD for the fossil-fueled plant with a natural-draft tower is at some point above 80°F as compared to the 55°-57°F range, and the optimum for a fossil-fueled plant with mechanical-draft tower is 78°F as compared to a 59°-61°F range. On the other hand, if a summer peak is assumed for Anchorage, Alaska, the optimum ITD for a fossil-fueled plant with natural-draft tower becomes 68°F as compared to a range of 80°F and above, and the optimum point for the mechanical-draft tower is 71°F as compared to a range of 79°-80°F.

SECTION XII

ECONOMIC COMPARISON OF THE DRY-TYPE AND THE EVAPORATIVE-TYPE COOLING SYSTEMS

In this report information has been presented as to the theory of dry-type cooling as it would apply to steam-electric generating plants; the operating results have been summarized for several existing dry cooling tower installations; the comments of equipment manufacturers have been summarized; and economic analyses have been made to indicate the economic factors which should be considered in the determination of the type of cooling system to be used for future generating plants.

To illustrate, the capital cost, based on 1970 prices, of a dry-type, mechanical-draft cooling system for use with a fossil-fueled generating plant plus the capital cost of the required supplemental peaking capacity would be approximately $25 per kw for the Chicago area, as indicated on Figure 55. In contrast, the cost for a wet-type, mechanical-draft cooling system may be on the order of $11 per kw and there would be no requirement for additional peaking capacity to provide comparable station output capacity. This would indicate a capital cost penalty for the dry tower installation of $14 per kw which is equivalent to approximately 0.34 mills per kwh based on a 15 percent fixed-charge rate, a 1 percent operating and maintenance charge, and the energy generation assumed for these analyses. The analyses of the dry cooling systems indicate that the higher operating back pressures of such systems coupled with the requirement for some energy generation by peaking units during periods of hot weather would result in an increase in annual fuel cost of approximately 1.8 percent as compared to a wet cooling tower installation. Assuming a fuel cost of 35¢ per million Btu and a plant heat rate of 9,000 Btu per kwh, the 1.8 percent difference is equivalent to about 0.06 mills per kwh. In addition, the fan power requirements of the dry cooling tower installation would be somewhat greater than for a wet tower and this may result in an additional penalty of about 0.08 mills per kwh for a total cost difference of about 0.48 mills per kwh. Table 34 lists a comparison of bus-bar costs of a dry-type versus evaporative-type cooling system. This cost difference, if not offset by some of the factors discussed below, is equivalent to about 7 to 10 percent of the cost of power and energy at the generating station bus bar. This is approximately 2 or 5 percent of the cost of power and energy at retail, reflecting all costs of generation, transmission and distribution. This indicates that, even without the benefit of potential cost savings discussed below, the impact of dry cooling on retail electric bills would be small. A 2 to 5 percent increase in a $20 monthly electric bill is equivalent to only 40¢ to $1.00 and an increase of even this magnitude would not occur unless all generating plants of a given utility were cooled by a dry-type cooling system.

TABLE 34

Comparison of Bus-Bar Costs
Dry-Type and Evaporative-Type Cooling Tower Systems
Mechanical-Draft, 800-Mw Fossil-Fueled Generating Unit
Chicago Area

	Mills per Kwh		
	Dry-Type System	Evaporative Type System	Difference
Plant Fuel Cost	3.210	3.153	.057
Cooling System Auxiliary Power Costs	0.140	0.062	.078
Cost of Capacity Replacement	0.193	0.000	.193
Cooling System Capital, Operation and Maintenance Costs	0.418	0.268	.150
		Net Diff:	.478

(1) The costs shown in this table reflect the study assumptions as summarized in Table 10.

(2) The annual average turbine heat rate with a dry-type tower is estimated to be approximately 1.8% higher than with an evaporative-type tower due to higher average back pressure operation, 9,170 Btu/kwh compared to 9,010 Btu/kwh. The above plant fuel costs reflect this difference.

(3) The mechanical-draft, dry-type cooling system capital cost used to develop annual costs in this table is $17.15/kw and the evaporative cooling system capital cost — $11.00/kw.

(4) The cost figures in this table are based upon a 15% fixed-charge rate, 35¢/10^6 Btu steam turbine fuel cost and 40¢/10^6 Btu gas turbine fuel cost.

(5) Weather data used is listed in Table A-VI of Appendix B.

(6) The evaporative-type system analyzed is based upon a range of 24°F, an approach of 18°F and a terminal temperature difference of 6°F.

(7) The dry-type system initial temperature difference is 61°F.

As shown above, the cost penalty of the dry-type cooling tower system is a small percentage of the total cost of power and energy delivered to the customer, even if no offsetting cost savings are assumed. It should be pointed out, however, that, since the dry cooling system requires very little water, an electric utility would have much more flexibility in locating its generating plant and would have the opportunity for certain cost savings. For example, a saving in fuel cost of approximately 5¢ per million Btu would entirely offset the cost difference of 0.48 mills per kwh computed above.

The greater flexibility which is afforded by the dry cooling system may make possible transmission savings which would offset a portion or all of the cost difference between the wet and dry systems, or may permit an additional generating unit to be built at an existing station even though there is not adequate water for an additional wet cooling tower. This would permit the utility to realize the economies of an additional unit at an existing facility.

The most obvious cost saving is that related to cooling water. If it is assumed that cooling water costs $100 per acre-foot (about 31¢ per thousand gallons), water cost savings, alone, for the dry tower installation would approximate 0.2 mills per kwh.

Based on the above, when all factors are considered, it appears that in many cases the dry cooling system would be economically competitive with wet cooling tower systems, and in some cases the dry system may have a decided economic advantage. The economic advantage would be most pronounced in those cases where the use of a dry-type cooling tower would allow the utilization of low-cost fuel in a water-short area.

In some cases, the relative economics of dry versus wet cooling will be overshadowed by pollution control considerations, and in these instances the dry cooling tower would, of course, have an advantage. The closed cycle of the dry cooling system means that thermal pollution of lakes, rivers, streams and the ocean from power plant waste heat would not occur. In addition, since there is no evaporation of water from the dry cooling system to increase the concentration of solids in the cooling water, there is no need for blowdown and, therefore, no danger of discharge of pollutants to the waterways.

SECTION XIII

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APPENDIX FOREWORD

This appendix contains background information and data which were used in the analyses made for this study.

Appendix A describes visits to existing steam-electric generating plants which are equipped with dry-type cooling towers to obtain first-hand information relative to the operation, maintenance and construction costs of these plants. Five stations equipped with dry-type cooling towers were visited—the Rugeley Station in England; the Ibbenbüren and Volkswagen plants in Germany; the Gyöngyös Station in Hungary, and the Neil Simpson Station in Wyoming. At the time of these visits, these stations had the largest electric utility operating units utilizing dry-type cooling towers. The operating experience gained from these existing cooling systems can be of value to those contemplating future installations.

Appendix B summarizes the ambient air temperature data utilized in the economic optimization analyses for the 27 United States sites considered, and refers to the source of this data. Temperatures were analyzed from $-40^{\circ}F$ to $+119^{\circ}F$ in 5° increments to determine their effect upon turbine back pressure and plant operating efficiency.

Important considerations to determine the economic choice of cooling system were summarized in Appendix C, "General Specifications for Dry-Type Cooling System Applications", as a guide for future installations.

Appendix D covers testing aspects for completed dry-type tower installations.

Procedures for developing cooling system costs used in the report are outlined in Appendix E.

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APPENDICES

Appendix A

Field Trips to Dry Cooling Tower Installations

RUGELEY STATION

Introduction

On December 15, 1969 John P. Rossie, accompanied by Mr. D. W. Crane, Project Engineer for English Electric Company, visited Rugeley Station. They were escorted by Mr. Platt, Assistant Superintendent of the station. At the time of the visit, Unit No. 3, which is equipped with the dry-type cooling tower, was in operation and carrying approximately 80 mw load. The air temperature was approximately 40°F and vacuum was approximately 1.5 inches Hg on Unit No. 3. Although Rugeley Station was operated as a base-load plant for the first few years after completion, it is now operated on a load-factor basis since larger, more efficient units operate base loaded. Unit No. 3 is called upon to operate at 120 mw during system peaks.

Description of Station

Rugeley Station of the Central Electricity Generating Board is located in the West Midlands Division adjacent to the Town of Rugeley, England. The original station, now designated as Rugeley Station "A", has a total generating capability of 600 mw, comprising five 120-mw units. All units are designed for an over-all thermal efficiency of 34.2 percent (9,980 Btu per kwh) and have throttle steam conditions of 1,500 psi, 1,000°F/1,000°F.

The station is at the site of the Lea Hall Colliery, which has been in operation for 600 years. Construction of Rugeley Station was started in July, 1955 and was completed in December, 1962. Since the completion of Rugeley Station "A", a new station—Rugeley "B", with two 500-mw units—has been constructed at the same site, but is not physically connected to the original plant.

With the exception of Unit No. 3 of Station "A", all the turbine-generators are equipped with surface condensers and evaporative-type cooling towers with reinforced, concrete, natural-draft towers of hyperbolic form. Make-up water for the evaporative-type cooling towers is pumped from the River Trent, which flows past the site.

The 120-mw Unit No. 3, which was commissioned in December, 1961 and has been in operation since, is equipped with a dry-type cooling tower of the Heller system design utilizing a concrete, hyperbolic, natural-draft tower. At the time of its construction, Rugeley Unit No. 3 was the largest Heller-type dry tower to be built. Up to that time, the largest such unit was a small pilot plant in Hungary.

Figure A1 (3A) illustrates the site layout of the station and Figure A2 is an aerial view of the plant. Note the difference in physical size of the dry tower on the right as compared to the size of conventional cooling towers serving units of the same mw rating. Approximately three times more air is moved through the dry tower than through each of the evaporative towers. The four evaporative-type cooling towers are each 350 feet high with base diameter of 216 feet. The dry-type cooling tower, which at the time of its construction was the largest concrete tower shell in the world, is 350 feet high and 325 feet in diameter at the base.

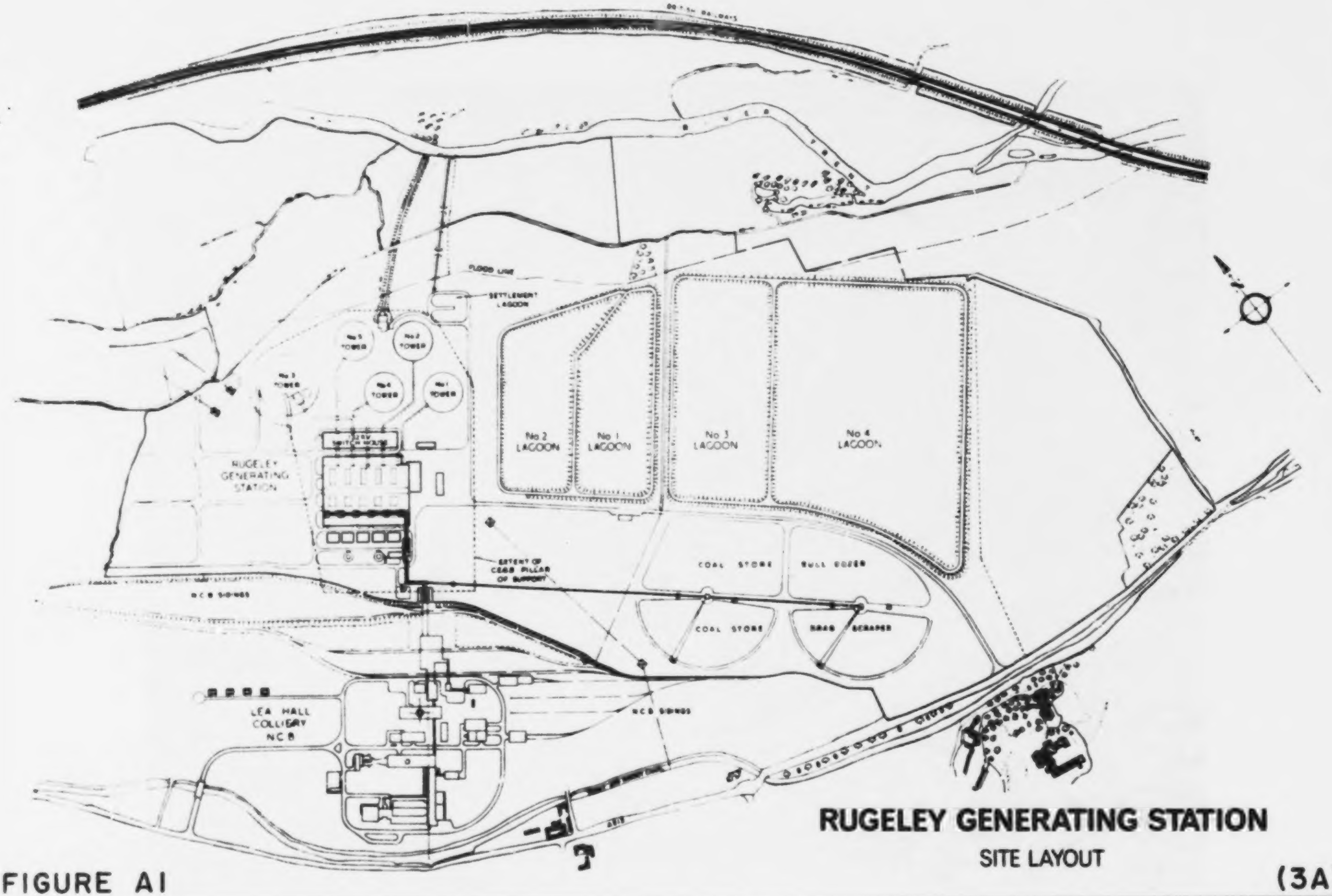
Water Circuit

Figure A3 shows a diagrammatic arrangement of the circuit in which 62,000 gpm of condensate quality water circulates (2A). There are four sectors in the cooling coils of the tower, each of which can be independently drained and filled with the other sectors in operation.

Two half-capacity circulating water pumps are provided to pump the water to the tower, and also to handle the small percentage of the flow which goes to the boiler feedwater system, amounting to approximately 3 percent of the total flow. The circulating water is conveyed to the tower through 60-inch-diameter pipes and is directed to each of the four equal cooling coil quadrants through specially designed sector valves.

From the sector valves, the water passes through the 48-foot-high columns of coolers. The Forgo coil used in the Heller system has a depth of six rows of tubes; water flow is upward in the inner three rows from the bottom of the column to the top, and the flow direction is reversed in the top water box downward through the outer three rows of tubes. Since the cooling air is flowing horizontally across the vertical tubes and comes into contact first with the lower temperature water, the system is designated as cross-counterflow.

After leaving the tower, the cooled water again passes through the sector valves and then through the two half-capacity recovery turbines which are connected to the same shaft as the circulating water pumps and motors. The purpose of the recovery turbines is to furnish a portion of the work necessary to drive the main



RUGELEY GENERATING STATION
SITE LAYOUT

FIGURE A1 (3A)

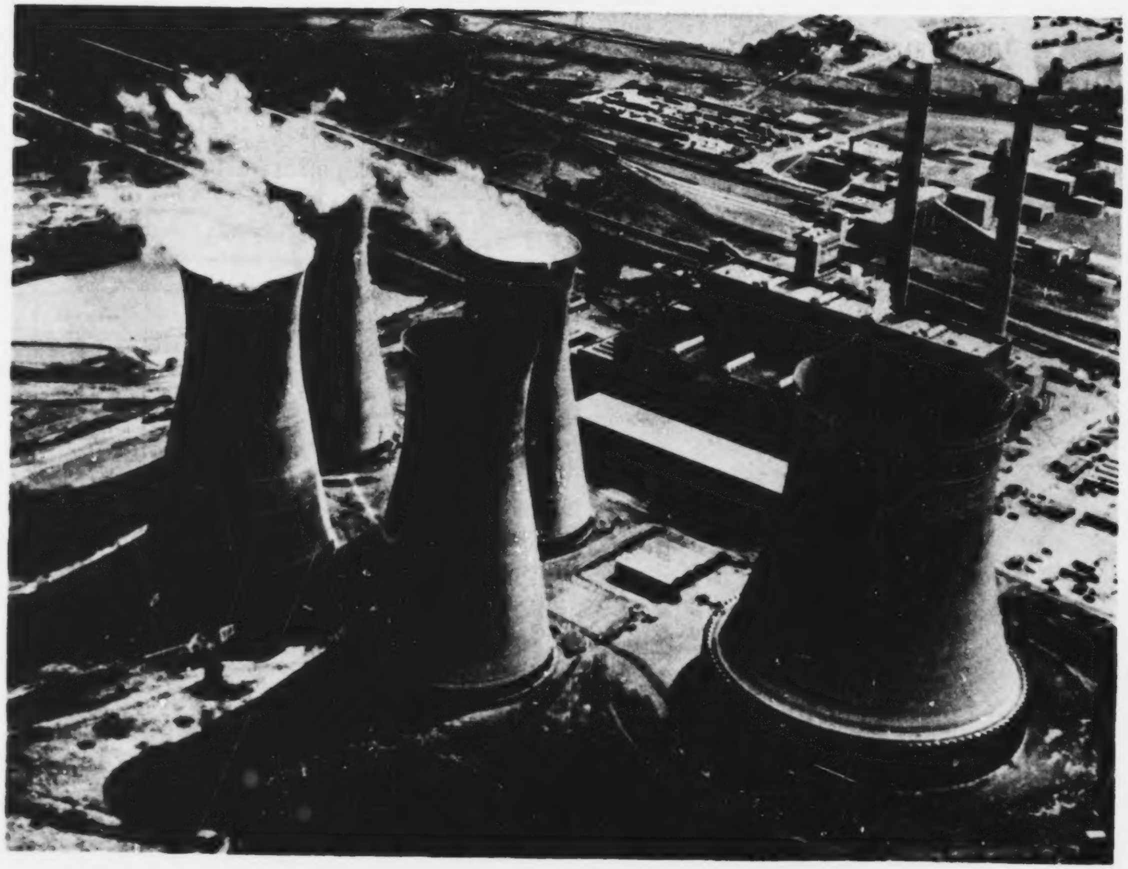


FIGURE A2— RUGELEY POWER STATION

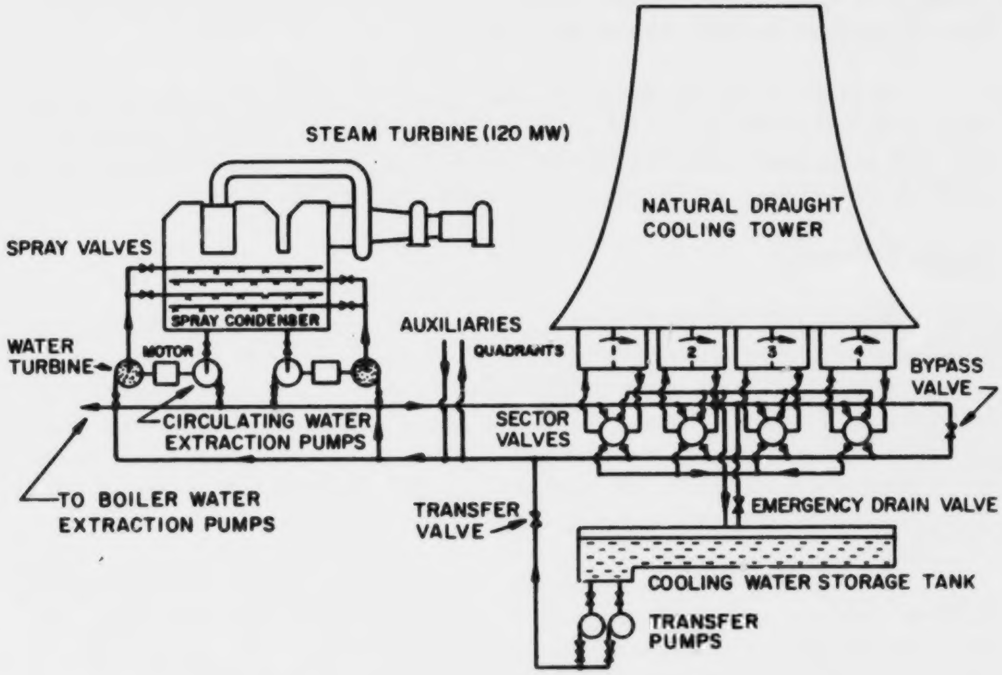


FIGURE A3— DIAGRAMMATIC ARRANGEMENT OF WATER CIRCUIT (2A)

circulating water pumps. An excess pressure of a few pounds per square inch is maintained at the top of the 45-foot cooling coil columns in order to have positive water pressure on the coils to prevent ingress of air in case of leaks. The recovery turbines utilize the major part of the excess pressure head to help drive the circulating water pumps.

From the recovery turbines, the water flows to the direct-contact condenser, through the spray nozzles where it condenses the exhaust steam from the turbine, and then is recycled through the cooling circuit.

In order to provide for quick drainage of the cooling circuit, a necessity in case operation of the unit is curtailed during freezing weather, an underground storage tank is located inside the base of the cooling tower. Two transfer pumps are used for transferring condensate and filling the coil sectors.

Design Parameters

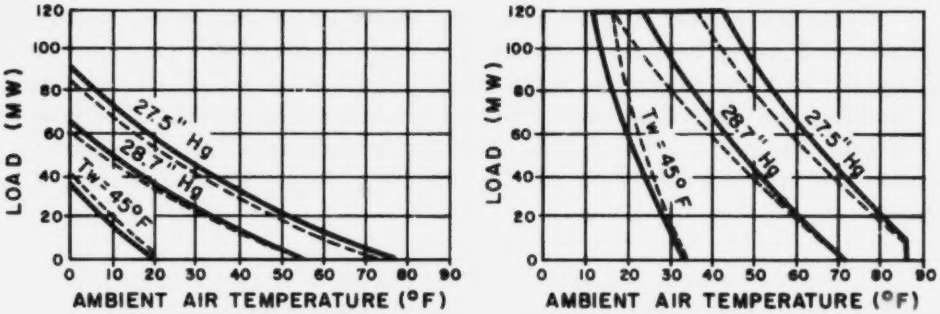
Apparently, the dry-type cooling tower was constructed at Rugeley for the purpose of obtaining experience with cooling towers which do not require a large amount of make-up water in anticipation of a shortage of water for power plant use in England. The desire to be able to construct power generating stations near a source of fuel or near a load center without being dependent upon an adequate supply of make-up water for a wet-type cooling tower was also a factor in the decision to obtain operating experience with a dry tower in England.

The turbine back pressure design at Rugeley No. 3 is 1.3 inches Hg with 52°F ambient air temperature, which is the same as the design of the other four 120-mw units equipped with evaporative-type cooling towers. The design initial temperature difference between the saturated steam temperature and the ambient air temperature is therefore 35°F, since the saturated steam temperature corresponding to an absolute pressure of 1.3 inches Hg is 87°F. The tower design heat rejection load is 587 million Btu per hour. Apparently the back pressure was not optimized but was selected so that generating plant equipment similar to the conventional 120-mw units could be utilized with the dry tower.

Figure A4 shows the design performance curves of the dry tower for one, two, three and four quadrant operation.

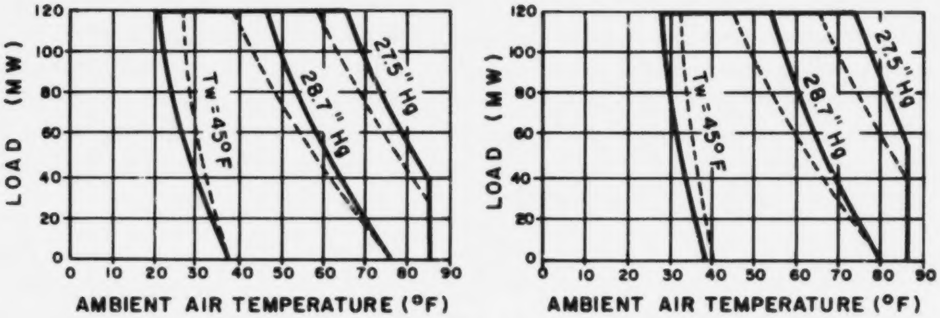
Capital Costs of the Dry Tower

No figures are available as to the construction costs of the dry tower system at Rugeley Station. However, representatives of the English Electric Company (the contractors for the equipment) advise that, in general, the components of a dry-type tower are from one and one-half to two times the cost of the components of an evaporative-type tower.



ONE QUADRANT OPERATION

TWO QUADRANT OPERATION



THREE QUADRANT OPERATION

FOUR QUADRANT OPERATION

NOTE:

SOLID LINES REFER TO OPERATION WITH TWO PUMPS AND DASHED LINES WITH ONE PUMP.

Tw = 45° F REFERS TO THE MINIMUM AVERAGE COOLER WATER OUTLET TEMPERATURE PERMITTED TO SAFEGUARD THE COOLERS FROM FREEZING.

FIGURE A4—DESIGN PERFORMANCE CHARTS FOR RUGELEY NATURAL-DRAFT COOLING TOWER (2A)

Manpower Requirements of the Tower

There are no special manpower requirements associated with operation of the dry tower at Rugeley because of the central control system which is installed with the tower. With the exception of the steam-jet air ejectors, all starting and control functions are carried out from the control room of the station. The time necessary to bring the dry tower system to operating vacuum level is approximately 10 minutes, which is the same time required for the units equipped with surface condensers.

There are no problems associated with establishing an air flow through the tower. Water is first circulated through the windward cooling section, or through two opposite sections if there is no wind. After the tower is in service, no operating functions are required unless the low-temperature alarm on the circulating water sounds, at which time the operator initiates the coil drainage sequence to take one or more cooling coil sections out of service. When conditions permit, the tower sections are returned to service from the control room. The process of removing sectors from service, restoring them to service and starting up and shutting down the dry tower system is accomplished by the automatic sequential control system which is initiated by control-button operation.

Winter Operation

Except for some minor instances of coils freezing as a result of automatic vent valves not operating, and faulty low-temperature alarms, operation of the dry tower during freezing weather has been satisfactory. Despite the freezing problems during the first winter's operation, Unit No. 3 was able to generate up to 137 mw in very severe weather at a time when the other four units with evaporative-type cooling towers were having operational difficulties because of tower icing.

Although the climate at Rugeley is not as severe as in continental Europe, temperatures below freezing are experienced regularly in winter. The lowest recorded temperature at Rugeley is 9°F.

Operation is controlled to keep the condensate temperature leaving the tower above 45°F as a precaution against freezing. Temperature control is obtained by taking cooling coil sectors out of service and draining the condensate in the idle section into the storage tank. Limited temperature control can also be achieved by taking one of the circulating pumps out of service. The operation of the drainage system to take cooling coil sectors out of service is initiated manually, and automatic sequential operation of valves and pumps follows.

The Rugeley tower is not equipped with louvers to control the flow of air across the coils.

Tests have been made which indicate that the coils can be safely filled and drained during freezing weather if the operation is accomplished within 2 minutes. The most critical operation is filling the coolers after a previous drainage when some ice has already formed in the tubes. Tests showed that water of 90°F inlet temperature would prevent freezing during filling with temperatures as low as −30°F and water of 100°F, down to −36°F. The procedure for placing the dry tower coolers into service during freezing weather is to bypass circulating water around the coils by means of the bypass valve until the water is heated up to 80° to 90°F before entering the cooling coils.

Description of System Components

Cooling coils. There are 648 cooling coils of the Forgó design in the tower. Each coil (element) is 16 feet high by 8 feet wide and 6 inches thick, containing 6 rows of 40 tubes in square pitch. The coils are arranged in columns, three elements high, with two columns joined together to form a "delta".

The cooling coils are constructed of aluminum which is 99.5 percent pure. The tubes, plate-type fins, and water boxes are all of aluminum construction. The total frontal area of the coils is 80,000 square feet.

Tower shell. The shell is hyperbolic in shape and is constructed of reinforced concrete of 5 inches minimum thickness, becoming thicker where a reinforced concrete ring beam takes the thrust. The tower is supported on reinforced concrete legs 55 feet high, which provide an opening for the air to pass through the coils and upward through the shell. Figure A5 shows a view inside the tower and shows the supporting structure and ring beam at the base of the tower. Tower dimensions are: height - 356 feet; base diameter - 325 feet; throat diameter - 205 feet.

Condenser. The condenser (Figure 33) is a direct-contact, spray-type designed by the English Electric Company and has a single steel shell mounted directly below the turbine receiving the exhaust steam from the 3-flow, low-pressure turbine cylinders. Cooling water is supplied through water boxes at each end of the condenser and is sprayed into the shell, mixing directly with the turbine exhaust steam. There are 24 spray pipes, fed alternately from opposite ends of the condenser through the water boxes. The spray nozzles are divided into four groups and each of the groups is fitted with a spray control valve of the butterfly type. The spray control valves are automatically controlled to close in case the circulating water pumps fail and there is a danger of flooding the condenser. The valve closure is automatically controlled to prevent water hammer damage to the piping system and coil sections. The original condenser design has been reworked to improve the performance because of subcooling of the condensate. When the unit was first placed into service, subcooling of the condensate as much as 15°F was experienced, which was found to be the result of air leakage and difficulty in removing

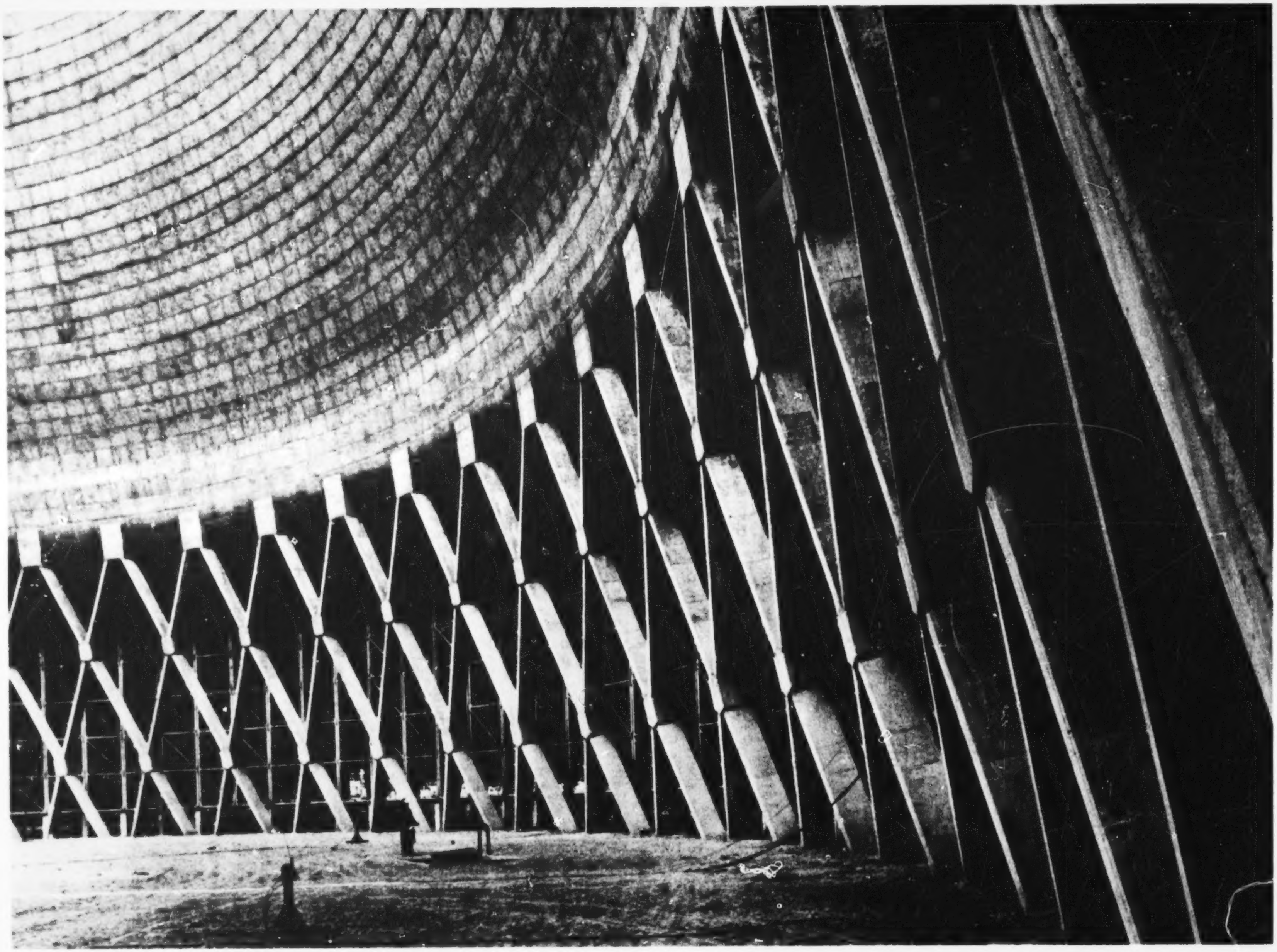


FIGURE A5—INSIDE VIEW, NATURAL-DRAFT, DRY-TYPE COOLING TOWER—RUGELEY STATION
(ENGLISH ELECTRIC PHOTO)

air from the condenser. By redesigning the air-collection system and rearranging some of the spray nozzles, the difficulties were eliminated and the subcooling was reduced to less than 1°F.

Sector valves. An interesting feature of the Rugeley Station is the use of sector valves to control the flow of circulating water. These valves are located in the pump house inside the tower and are of the multi-port rotary plug design. The sector valves are used for isolating individual cooling sectors of the tower, for filling and draining sectors and for normal operation.

The sector valves were designed especially for the Rugeley dry tower. The other stations using Heller-type towers which were visited did not use this type of valve, but relied upon individual valves to perform the various functions.

Figure A6 shows a view of the sector valves in the pump house.

Auxiliary Power Requirements

The total auxiliary power requirements for Unit No. 3 are 8.9 mw at full load, or approximately 7.3 percent. The power-using auxiliaries associated with the operation of the dry tower are the two half-capacity circulating water pumps, the power use of which is compensated for in part by the energy regained by the water-recovery turbines. The main circulating pumps are each 1,104 kw and the recovery turbines, 324 horsepower (242 kw). The net pumping requirement at full load with both pump and recovery turbines in operation is approximately 1,723 kw, or 1.4 percent of output. The recovery turbines recover approximately 22 percent of the pumping power.

Cooling water for the generators, oil coolers, bearing service and other auxiliary cooling requirements is furnished by a small auxiliary or dry-type tower which uses mechanical draft for moving the air across the coils.

Turbine Cycle Performance

The turbine cycle design heat rate for Unit No. 3 at Rugeley is the same as for the other four units which are equipped with evaporative towers (3A). The turbine cycle efficiency of all units is 41.3 percent, equivalent to 8,264 Btu per kwh with 1.3 inches Hg back pressure. Because of the higher back pressure actually experienced with Unit No. 3, station records made available by the station superintendent indicate rhe turbine cycle heat rate is slightly higher than design.

However, as reported in (1A), the performance at design point (vacuum 28.7 inches Hg at 120 mw and air temperature 52°F) has been met, and performance throughout the operating range closely follows that predicted. Table A-I shows

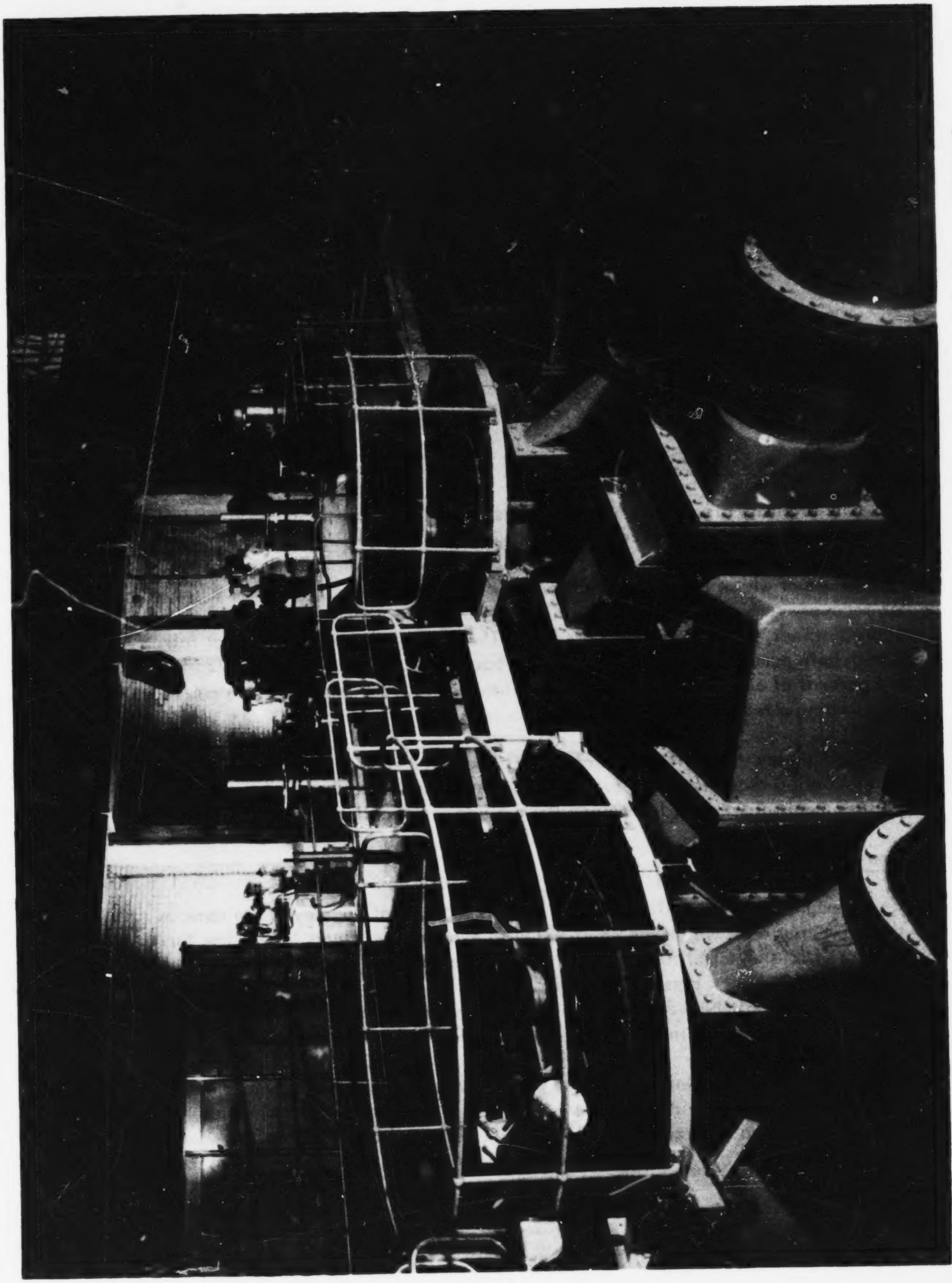


FIGURE A6—SECTOR VALVES, VALVE HOUSE LOCATED INSIDE NATURAL-DRAFT, DRY-TYPE COOLING TOWER—RUGELEY STATION (ENGLISH ELECTRIC PHOTO)

TABLE A-1

Operating Data – Rugeley Power Station
120-Mw Turbine-Generator — Unit No. 3
with Natural-Draft Dry Cooling Tower

Ambient Air Temperature (°F)	Wind Velocity (mph)	Condenser Loading (lbs. of steam/hr.)	Back Pressure (in. Hg)	Auxiliary Power for Cooling System (mw)	Net Output (mw)
43.3	12 N.E.	700,000	1.85	1.7	111.1
51.4	18 N.W.	700,000	2.62	1.7	111.1
41.0	18 N.	700,000	1.71	1.7	111.1
36.0	4 S.W.	700,000	1.52	1.7	111.1
29.8	8 S.E.	700,000	1.55	1.7	111.1
22.5	8 N.	700,000	1.31	1.7	111.1
26.2	25 N.E./N.W.	700,000	1.37	1.7	111.1
32.6	10 E.	700,000	1.53	1.7	111.1
38.0	20 E.	700,000	1.69	1.7	111.1
27.9	10 W./N.E.	700,000	1.50	1.7	111.1

operating results as logged by station operating personnel with 10 operating conditions selected at random from a period of time from November, 1966 through February, 1969.

Corrosion Problems

The Rugeley Station is located at the site of a coal mine and is also adjacent to an ash-sintering plant which makes building blocks from the plant ash. Also, the dry-type cooling tower is in close proximity to the four evaporative-type towers and subject to any drift of spray from these towers.

Within a short time after start-up, serious corrosion was found to have started in the crevices between the cooling coil fins and the spacer collars of the cooling sections, and also in the tube walls beneath the spacer collars. The Forgó coil is constructed by placing an aluminum collar over each aluminum tube, followed by a section of the aluminum plate fin. The collars and fins are stacked alternately on the tubes until each coil is completed, at which time the fins and collars are tightly pushed together by a hydraulic press; then an expanding mandrel is drawn through the tubes, resulting in a tight mechanical bond between tubes, collars and fins.

Apparently, the combination of moisture in the air and pollutants, especially chlorides, was able to find its way into the tiny crevices despite the tight mechanical joint between the fins and collars, setting up corrosion cells.

Based upon experience gained in Hungary with the Heller system, no corrosion was expected at the Rugeley plant. However, the corrosion advanced to the point where tube walls were perforated. Also, the products of corrosion which were deposited in the fins caused damage to the fin surfaces. Damage was more severe on coil sectors which were out of service.

A program of research was undertaken by the Central Electricity Generating Board and the English Electric Company and a protective coating of epoxy resin was selected as the best method of corrosion prevention. At the present time, a large number of the cooling coils at the Rugeley Station have been treated with the epoxy coating.

Effect of Wind on Tower Performance

Since winds have an adverse effect upon the performance of a natural-draft cooling tower (both the evaporative type and the dry type), tests were made at Rugeley to measure the effect of the wind.

With wind velocity up to 10 mph, no effects were noticed, but loss of vacuum was observed at wind speeds above 10 mph, with a vacuum deterioration of approximately 0.36 inches Hg for a wind speed of 30 to 35 mph. The average annual wind speed at Rugeley is 13 mph; thus, the effect of wind on tower performance is not a significant factor.

The air flow around the cylindrical tower causes a reduction in air flow through the coolers. Figure A7 shows the effect of wind on the tower performance as observed during the tests.

Other tests were conducted to determine the variation of air mass velocity through the coolers around the tower. Figure A8 shows the results of tests made at full load and wind of 20 mph. The terms "upstream" and "downstream" refer to the position of the coolers in the deltas, or V-shaped sections of coolers. The survey revealed that the loss of vacuum was mainly a result of the blanketing of the downstream deltas of those coolers having tangential wind components, which more than offset the increased air flow through the upstream coolers (1A).

Other effects of weather which have been observed are: fog improves performance; rain reduces performance slightly; and intermittent sun produces a flicker on the vacuum gauge.

Water-Side Chemistry

The high purity aluminum tubes used in the cooling coils require close control of the pH of the circulating water. In order to prevent corrosion of the aluminum water-side surfaces and to keep the aluminum from going into solution in the water and ultimately depositing in the turbine blades, a lower pH is carried in the tower circuit than in the boiler feedwater circuit of Unit No. 3.

In order to protect the steel surfaces of the circulating water system piping from the low pH of the water, the inner surfaces were coated with plastic.

Both the aluminum and iron content of the circulating water has remained satisfactory; aluminum 0.01 to 0.02 ppm and soluble iron 0.02 ppm. The pH of the boiler feedwater is controlled by morphaline. The dissolved oxygen content of the water in the tower circuit is 0.1 to 0.3 ppm. Oxygen content of the boiler feedwater after the deaerating feedwater heater is reported to be about 0.02 ppm, which is considered satisfactory. Neither the boiler nor the turbine have experienced deposits.

Maintenance

Other than the repairs and cooling coil coating with epoxy which was necessitated by the external corrosion, there are no extraordinary maintenance

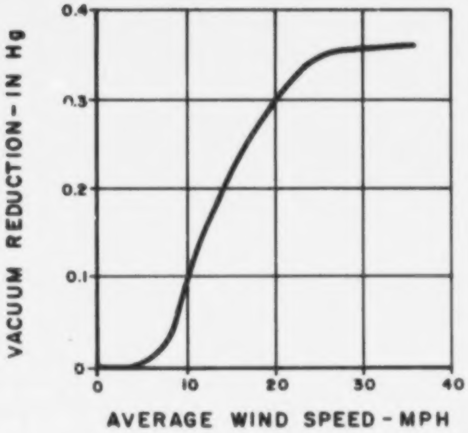


FIGURE A7— WIND EFFECT UPON PERFORMANCE (IA)

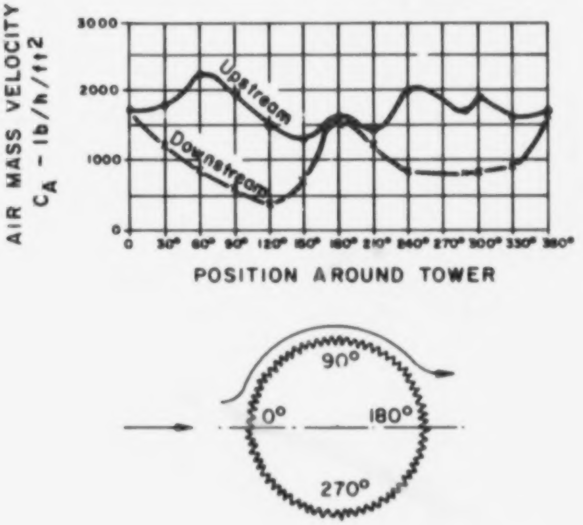


FIGURE A8—AIR MASS VELOCITY VARIATION THROUGH THE COOLERS WITH A 20 M.P.H. WIND (IA)

problems associated with the dry tower. The station superintendent at Rugeley, Mr. J. E. Farrington, reported that dry tower maintenance has been relatively low, disregarding the corrosion of the cooling elements, and is mainly associated with venting valves and quadrant valves.

It is reported that no cleaning of the coils has been necessary to remove dirt and soot. A thin deposit forms on the exterior surfaces of the coils and fins and reaches equilibrium with minor influence on performance.

Conclusion

Although exterior corrosion has been a major problem, the Rugeley dry tower is considered a success in that much useful information was gained towards advancing the art of dry tower design, construction and operation.

IBBENBÜREN PLANT

Introduction

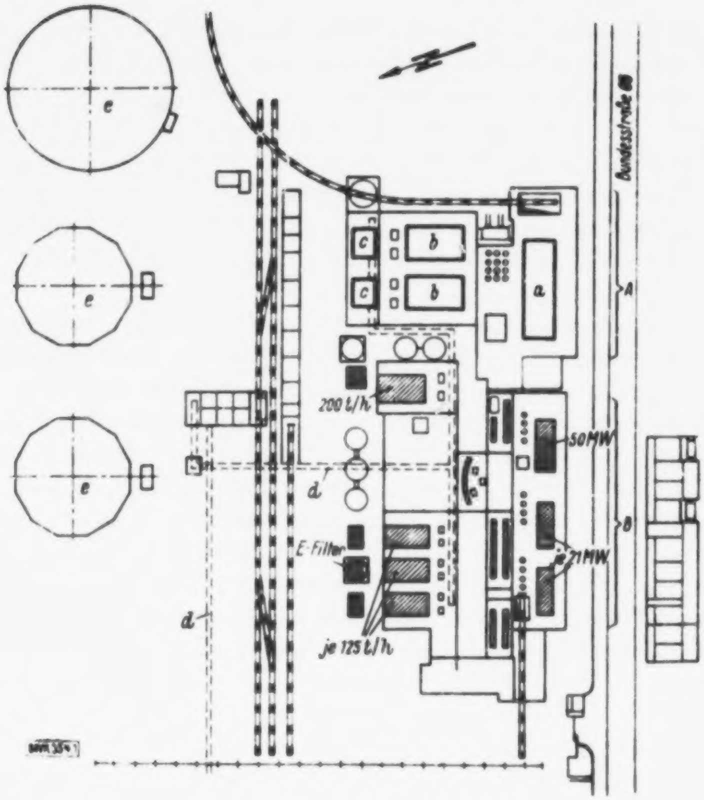
On Wednesday, December 3, 1969, John P. Rossie, accompanied by Mr. Hans-Bernd Gerz of the Development Division of the Firm of GEA (Gesellschaft für Luftkondensation) of Bochum, West Germany, visited the Ibbenbüren power plant of the Preussag AG in Ibbenbüren, West Germany. The Ibbenbüren plant is equipped with a Heller dry-type cooling tower and the equipment for the plant was supplied by GEA. During the visit to the plant, the party met with Mr. Ottokar Scherf, Superintendent of the plant, and was escorted around the plant by his assistant, Mr. Hoffmann. At the time of the visit, the unit equipped with the dry-type cooling tower was operating at design load of 150 mw. The turbine back pressure was 1.75 inches Hg and the ambient air temperature was 39°F.

Description of Plant

The Ibbenbüren plant is located in the Town of Ibbenbüren in the Ruhr Valley, the area where much of the heavy industry of West Germany is concentrated. The plant is located at the site of an underground coal mine in an area where mining has been undertaken for over 500 years. Preussag, the corporation which owns and operates the plant, is engaged primarily in coal mining operations.

The coal mined at the Ibbenbüren plant is anthracite, hard coal and forge coal; over 2 million tons of coal per year are mined. Much of the coal is for home fuel, but the fines and smaller granulated coal are sold to industrial plants and power plants and shipped via railroad transportation. However, there is a certain amount of the coal which is high in ash and moisture and is not considered suitable for sale. In order to utilize the low-grade coal at the site, a power plant was constructed by Preussag. This plant went into operation in 1954. The original plant, of 100-mw capability, is served by two natural-draft, evaporative-type cooling towers of concrete construction and hyperbolic shape. The original plant consists of four boilers, three of which have a capacity of 275,000 pounds of steam per hour and one boiler with a capacity of 400,000 pounds of steam per hour, with 1,100 psi pressure and 968°F serving two 21-mw automatic extraction turbine-generators and one 50-mw regenerative cycle condensing turbine-generator. Figure A9 illustrates the Ibbenbüren plant layout.

The electrical output of the plant supplies the energy requirement of the mining operation; however, the greater part of the production is sold via the German electrical grid to various utilities.



a MACHINE HOUSE (150 MW)
b BOILER HOUSE (573,200 #/HR)
c ELECTRO FILTER
d CONVEYOR BRIDGE
e COOLING TOWER
A. NEW UNIT (150 MW)
B OLD POWER STATION (APPROX. 100 MW)

FIGURE A9 — PLAN VIEW OF PREUSSAG POWER STATION IBBENBÜREN (5A)

As coal production increased and more high-ash coal became available, Preussag decided to construct an addition to the generating plant to utilize the additional low-grade coal. In 1967, a 150-mw unit equipped with a Heller dry-type cooling tower was placed into service. Two 600,000-pound per hour capacity boilers with steam conditions of 2,700 psi, 977°F/977°F were installed to serve a 150-mw reheat turbine-generator. The tower is reinforced concrete, natural-draft type of hyperbolic shape.

The lack of suitable water for cooling tower make-up at a sufficiently low price was the reason for selecting a dry-type tower for the 150-mw unit at Ibbenbüren. The existing water plant has a maximum daily capacity of approximately 4 million gallons per day. The wet-type cooling tower of the original 100-mw plant, alone, uses 2 million gallons per day of this supply; thus, it would have been necessary to construct additional water-treating facilities and develop a new water supply if an evaporative cooling tower were chosen, since the wet tower would require an additional water make-up of approximately 3 million gallons per day.

Engineering studies were made by Preussag to compare the economics of a generating unit equipped with a dry-type cooling tower, which would require no make-up water, and a generating unit with a conventional evaporative tower. Consideration was given to the difference in initial construction cost, the cost of water, differences in operating efficiency because of higher back pressure with the dry tower, and other pertinent factors. From these studies, Preussag concluded that the price of make-up water would have to be lower than the range of 27¢ to 33¢ per thousand gallons in order for the total annual operating costs of a wet tower to equal that of a dry tower. This compared with the actual price of water of 47¢ per thousand gallons from Preussag's water-treatment plant, and 65¢ per thousand gallons from an outside supply of water which would have had to be developed for a new evaporative tower.

In performing the studies, Preussag investigated the Heller (indirect) system, the direct, air-cooled, condensing system and a combination of various cooling systems before selecting the Heller system.

Water Circuit

Figure A10 shows the diagram of the water circuit through which approximately 66,000 gpm of condensate quality water is pumped. The cooling tower is divided into four sections, each of which can be independently drained and filled.

The water is circulated through the tower by means of two half-capacity motor-driven circulating water pumps which are also equipped with Francis-type recovery turbines.

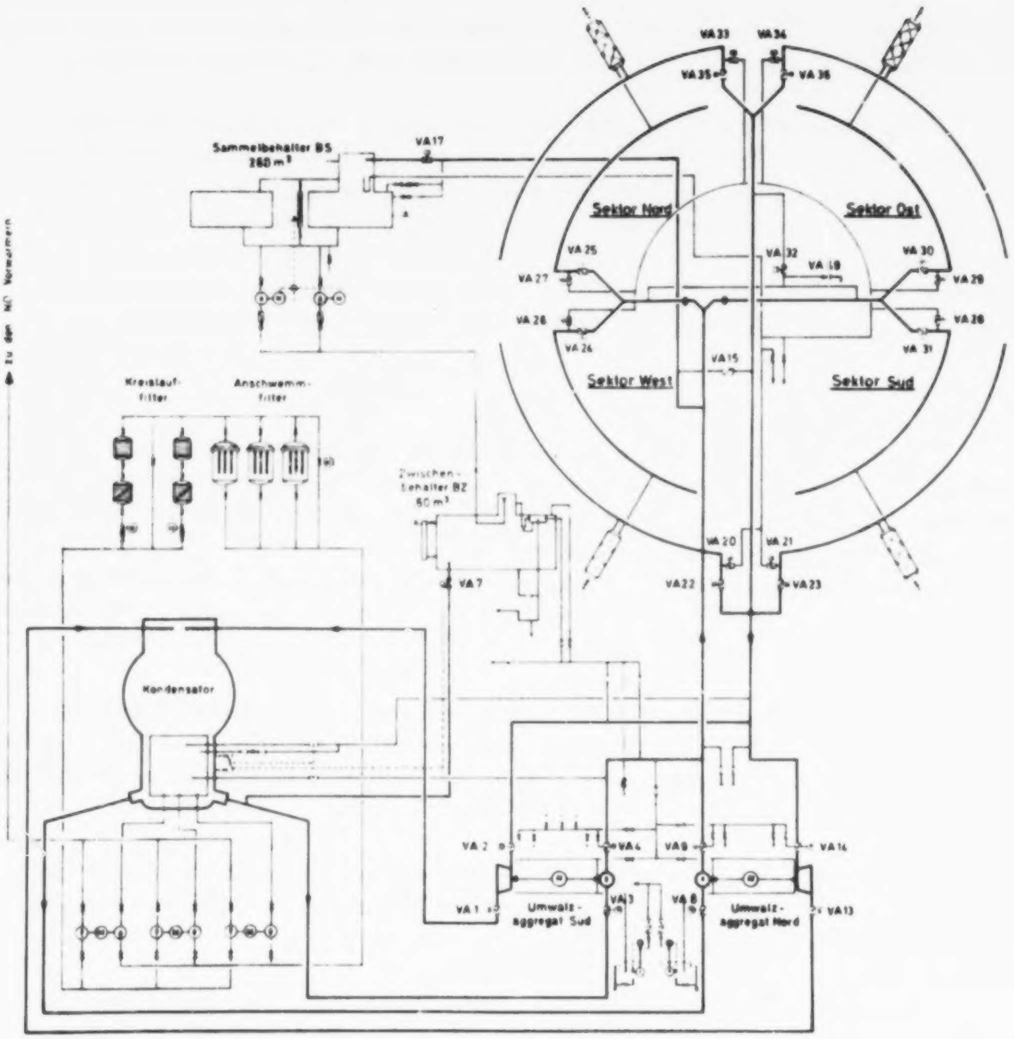


FIGURE A10 — COOLING WATER CIRCUIT DIAGRAM — 150 MW IBBENBÜREN GENERATING STATION (5A)

The Heller coils are of the same general construction and arrangement as at the Rugeley plant, with 48-foot-high columns of coils joined into deltas and operating in cross-counterflow pattern. However, sector valves are not used to control water flow. Instead, motor-operated butterfly valves direct the water flow for regular operation, drainage and filling.

Two underground water-storage tanks are provided inside the tower base. One is sized to hold the water in one sector and the other can hold the water from the entire system. Two filling pumps are provided with the storage reservoirs.

The direct-contact condenser is of somewhat different design than the Rugeley condenser. Rather than the circulating water being sprayed from the individual water spray pipes supplied from water boxes, the circulating water flows through four large water chambers in the condenser and 2,600 water spray valves are connected directly to the distribution chambers. Subcooling is reported to be less than 1°C. The water-pressure drop in the condenser nozzles is approximately 8 feet. Figure A11 shows a cross section of the direct-contact condenser.

Design Parameters

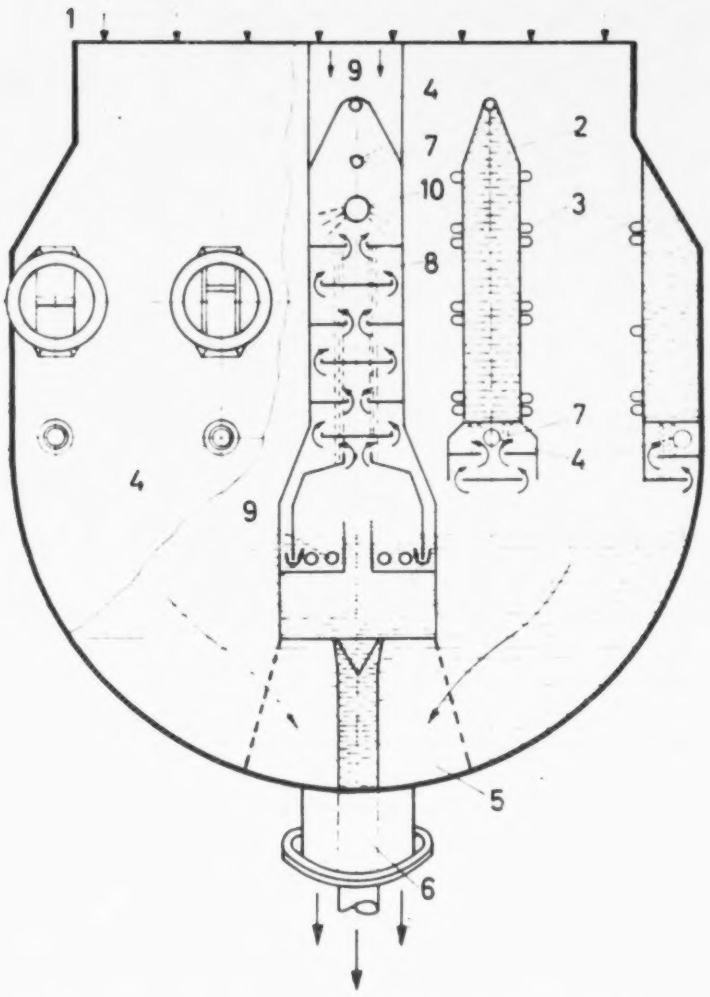
The turbine back pressure design of the Ibbenbüren plant is 1.22 inches Hg with 34.7°F ambient air temperature and a tower heat rejection of 645 million Btu per hour. The design initial temperature difference, which is the most important criterion affecting tower performance and initial cost, is 50.5°F. This compares to the Rugeley tower design of 1.3 inches Hg with 52°F ambient air and an initial temperature difference of 35°F, which resulted in a greater tower surface in proportion to the heat rejection load for Rugeley.

The Ibbenbüren tower has 498 cooling elements as compared to 648 for Rugeley, which reflects the higher back-pressure design resulting from the optimization studies made by Preussag before specifying the design parameters of the dry-type cooling tower.

Figure A12 shows the operating characteristics of the Ibbenbüren dry tower at various loads and ambient air temperatures. This curve was replotted from its original version (5A) to English units.

It is interesting to note that the cooling tower coils were formed into 83 delta sections at the factory in Jaszbereny, Hungary and transported by rail to Ibbenbüren without damage. A repetition of the pressure test at the site showed no leaks.

The Ibbenbüren tower is equipped with horizontal air-control shutters. Some 90 percent of the shutters are operated by electric motors controlled from the central



1. STEAM INLET
2. COOLING WATER INLET
3. SPRAY JETS
4. AIR EXTRACTION
5. COOLING WATER OUTLET
6. FEED WATER OUTLET
7. FINAL COOLER
8. DEAERATOR
9. INLET FOR DEAERATING STEAM
10. CONDENSATE INLET TO DEAERATOR

FIGURE A11 — DIRECT CONTACT CONDENSER (5A)

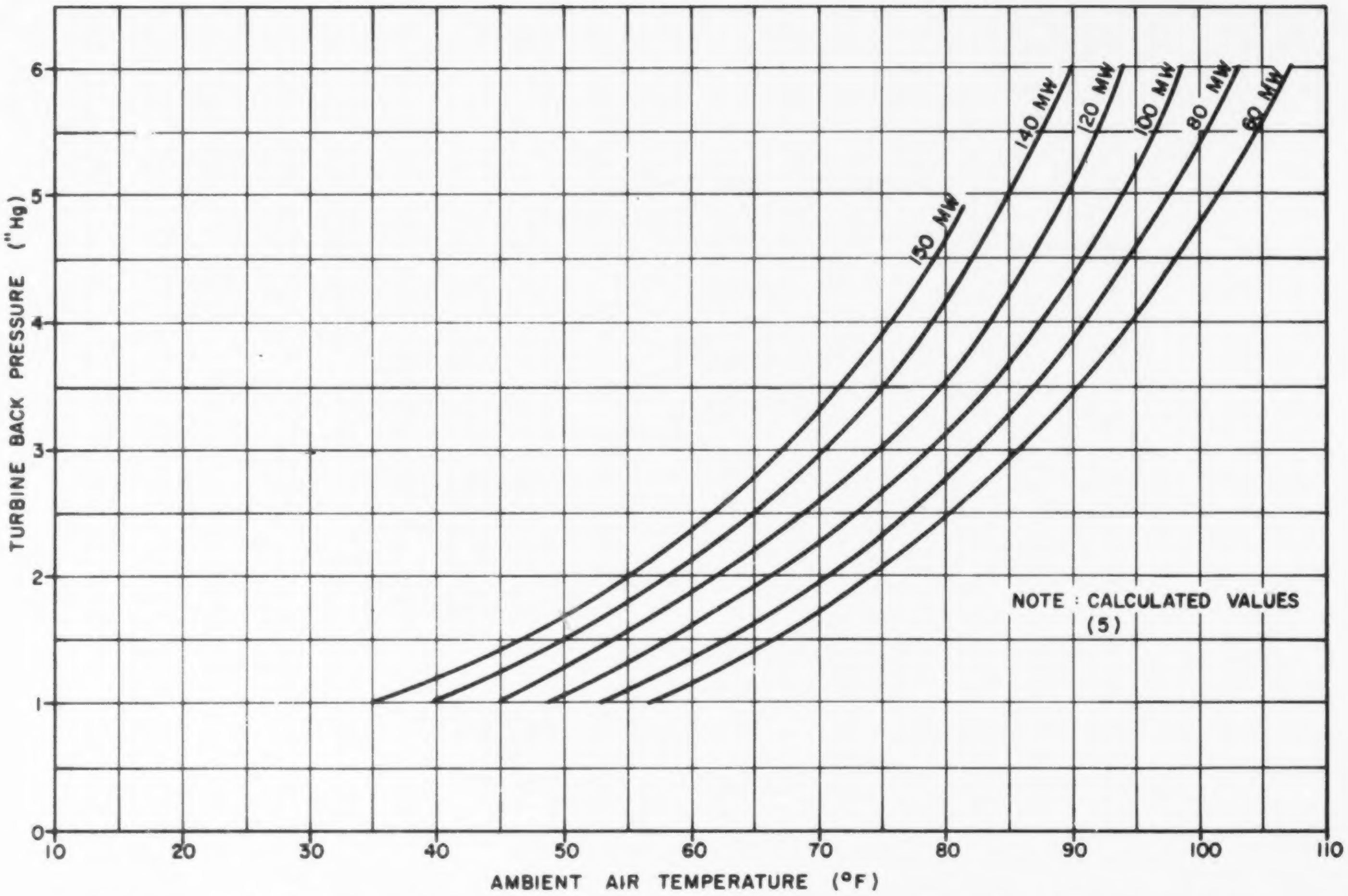


FIGURE A12 — IBBENBUREN PLANT—NATURAL-DRAFT, DRY-TYPE COOLING TOWER
TURBINE BACK PRESSURE VARIATION WITH AMBIENT AIR TEMPERATURE

control room by pushbuttons and the remainder are controlled manually. Figure A13 shows the Ibbenbüren tower with the shutters on the outside of the cooling coils.

Capital Costs

Mr. Scherf reported that the capital cost difference between the dry tower installation and a comparable wet tower installation was estimated to be approximately $1,500,000, noting that almost $400,000 in water costs are saved annually by the dry tower. The total cost of the plant, including the dry tower, was $22,800,000, making the cost of the dry tower plant approximately 7 percent higher than a plant with a conventional tower.

Manpower Requirements of the Tower

During the planning of the 150-mw unit, special attention was given to limiting the number of operating personnel required for the expanded plant. A criterion was adopted that no more operators should be utilized with the new addition than were required for a conventional wet tower installation. To accomplish this purpose, the components of the dry tower system were equipped for centralized control and automated to a great extent. All selections of pumps, draining and filling of coil sectors, shutter operation, and valve operation are performed from the central control room.

The 150-mw unit utilizes 8 men per shift as follows:

1	Shift Foreman
1	Turbine and Tower Control Operator
2	Boiler Control Operators
1	Roving operator who watches machinery, including the tower and condenser
1	Turbine-driven Boiler Feed Pump Operator
2	Boiler Auxiliary Operators and Ash Transporters

Only one visit per day is made to the tower by the operator.

Figure A14 shows the type of instrument which is used by the tower operator to determine when to shut down or restore a circulating pump to service and when to operate louvers.

By observing in which zone of the indicator the mw pointer and the ambient air temperature pointer cross, the operator is alerted as to when to operate louvers or circulating pumps to keep condensate temperature up to a safe level.

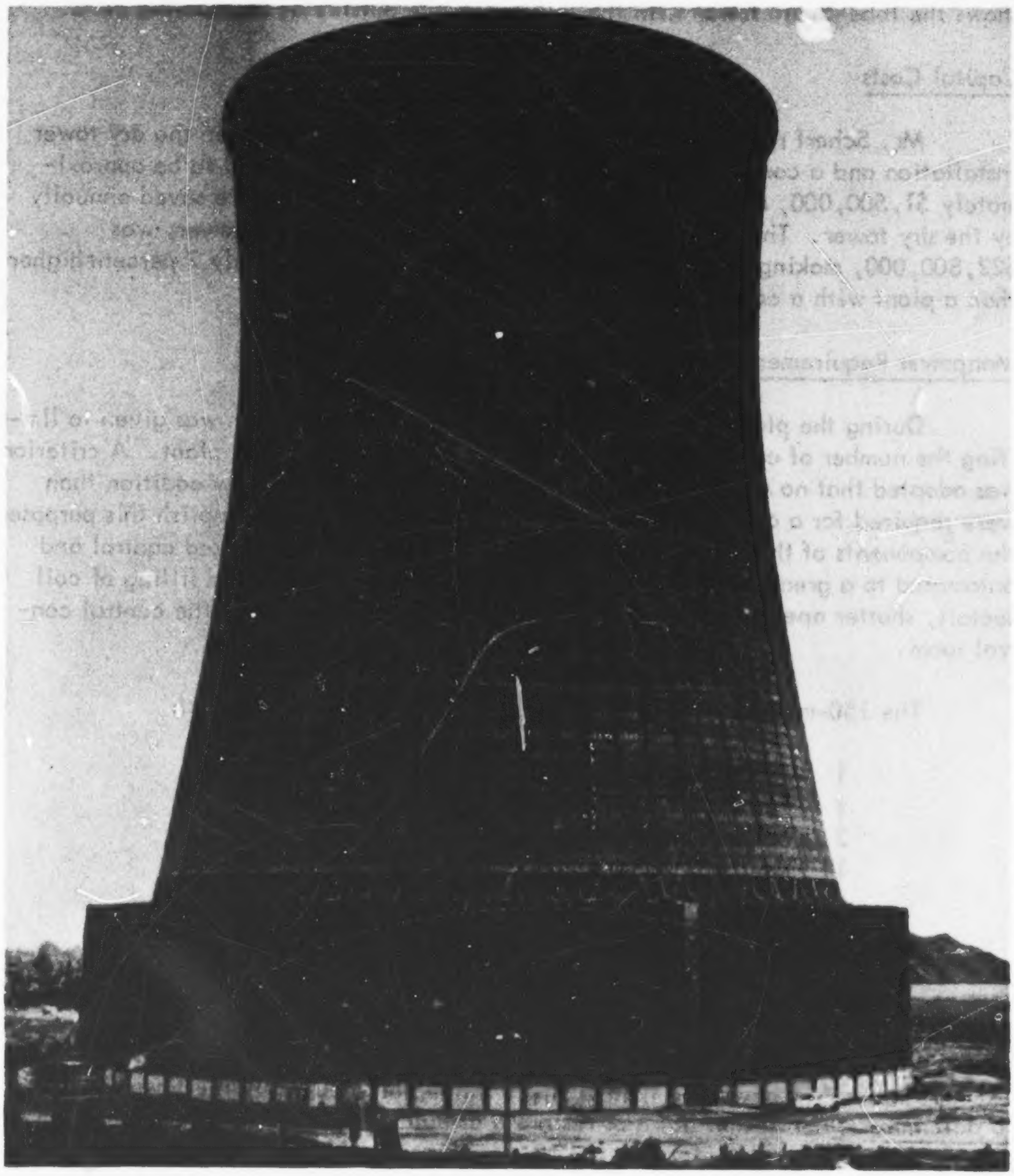
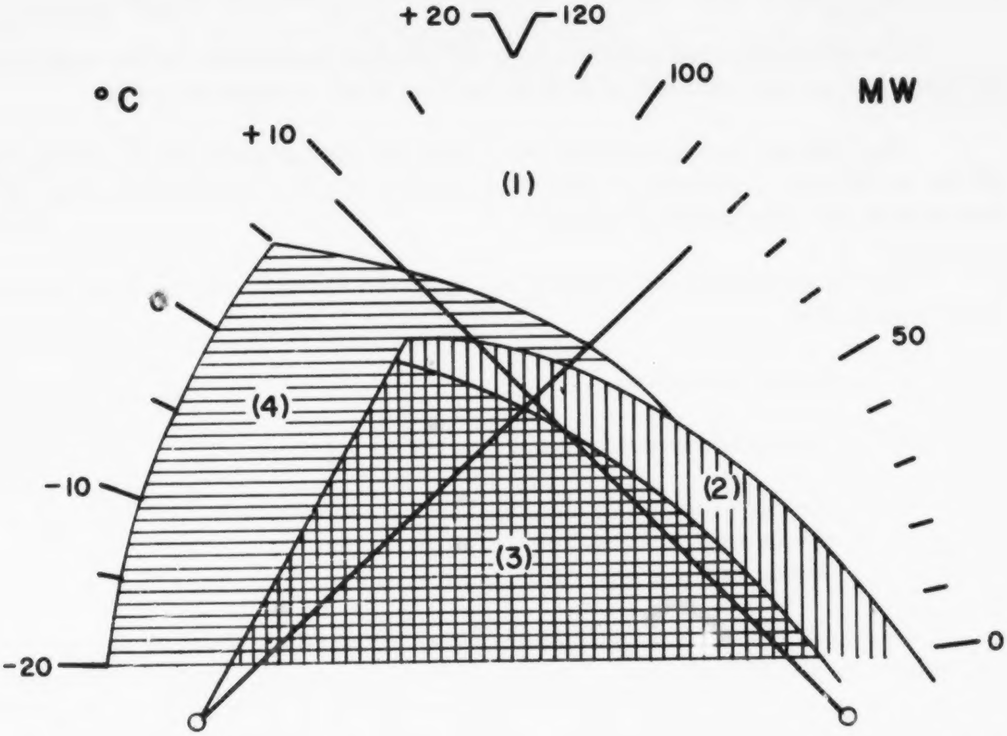


FIGURE A13— HYPERBOLIC CONCRETE DRY-TYPE COOLING TOWER INSTALLATION AT IBBENBÜREN - 150 MW GENERATING PLANT (GEA PHOTO)



(1). BOTH PUMPS RUNNING, LOUVERS OPEN
(2). ONE PUMP RUNNING, LOUVERS OPEN
(3). ONE PUMP RUNNING, LOUVERS CLOSED
(4). BOTH PUMPS RUNNING, LOUVERS CLOSED

FIGURE A14—OPERATIONAL CONTROL INSTRUMENT (5A)

Winter Operation

The Ibbenbüren tower differs from the Rugeley installation in that remotely operated louvers are installed to control the flow of air through the coils.

The 150-mw unit operates on the system load factor with output varying from 30 mw to 150 mw. Generally, the unit is held at rated load all day and drops off in load during the night and on weekends.

The means available to control the operation of the tower in winter to prevent freezing are:

1. Louver operation.
2. Taking one circulating pump out of service.
3. Draining of cooling sectors when the condensate temperature falls below a level where there is danger of freezing.

Drainage of sectors is initiated by a pushbutton with automatic sequential operation after the initial signal. It was reported that during the coldest weather and at loads as low as 30 mw, it has not been necessary to drain a sector to prevent freezing of the coils during operation. Apparently, at loads below 30 mw, it would be necessary to drain sectors during freezing weather. The lowest temperature recorded at Ibbenbüren is −4°F. The 50-year average temperature is 48°F and there is an average of 30 hours per year with air temperatures above 77°F.

During the initial winter operation, coils were damaged by freezing when an automatic vent valve failed to open to allow the column to drain to the storage tank. The trouble was corrected by replacing the vent valves by a new design insulating and protecting them from freezing with electric heating cable. The cooling coil column was replaced and no further trouble was encountered with freezing.

In placing the tower into service during freezing weather, the condensate from the condenser is recirculated, bypassing the cooling towers until the water is heated to a temperature high enough to safely fill the coils. The operation of filling the sectors is done automatically by the control system with sequential interlocks.

Draining of the tower is accomplished in 22 seconds and filling in 5 minutes.

Auxiliary Power Requirements

The total auxiliary power requirements of the 150-mw Ibbenbüren generating unit are approximately 8 percent of generator output.

The auxiliary power-using equipment associated with the dry-type tower are the two circulating pumps, which are equipped with water-recovery turbines to recover the excess head imposed upon the cooling coils and to maintain approximately 3 psi pressure at the top of the coils. The reason for maintaining the excess pressure is to have a positive pressure on all parts of the large area of cooling coils so that, in case of coil leaks, air will not be drawn into the system.

The total pumping power required by the circulating pumps is 1,640 kw, which is reduced by 550 kw recovered in the water turbines, for a net pumping power requirement of 1,090 kw, or 0.72 percent of the plant output. The water turbines recover 33 percent of the pumping power.

Turbine Cycle Performance

Since the steam conditions and heater cycle of the old and new sections of the Ibbenbüren plant are different, it is not possible to make a direct comparison between the units served by a wet tower and those served by a dry tower.

The design heat rate for the new unit is 8,400 Btu per kwh (2,100 K calories per kwh) at 1.2 inches Hg as compared to 11,500 Btu per kwh (2,900 K calories per kwh) for the older low-pressure units without reheat.

Corrosion Problems

Contrary to the experience at Rugeley Station, no external corrosion problems with the cooling coils have been experienced at Ibbenbüren.

As at Rugeley, the Ibbenbüren dry tower is located next to a wet tower and, presumably, also subject to drift of water spray from that source, although no special coating on the fins or tubes was applied at Ibbenbüren.

Effect of Wind on Performance

The same adverse effect of the wind noted at the Rugeley Station was observed at Ibbenbüren, with the exception that tests made at Ibbenbüren indicate that wind speeds as low as 2.24 mph (1 meter per second) influenced tower performance, whereas the Rugeley tower was not influenced up to 10 mph.

Tests made at Ibbenburen to verify tower performance show that for 6.7 mph (3 meters per second) wind velocity, the cooling effect is reduced by 2.7°F (1.5°C) and for 9 mph (4 meters per second) the cooling effect is reduced by 5.5°F (3°C).

Figure A15 shows a curve of the deviation of cold water temperature from that obtained under ideal conditions (optimum performance of the cooling tower at

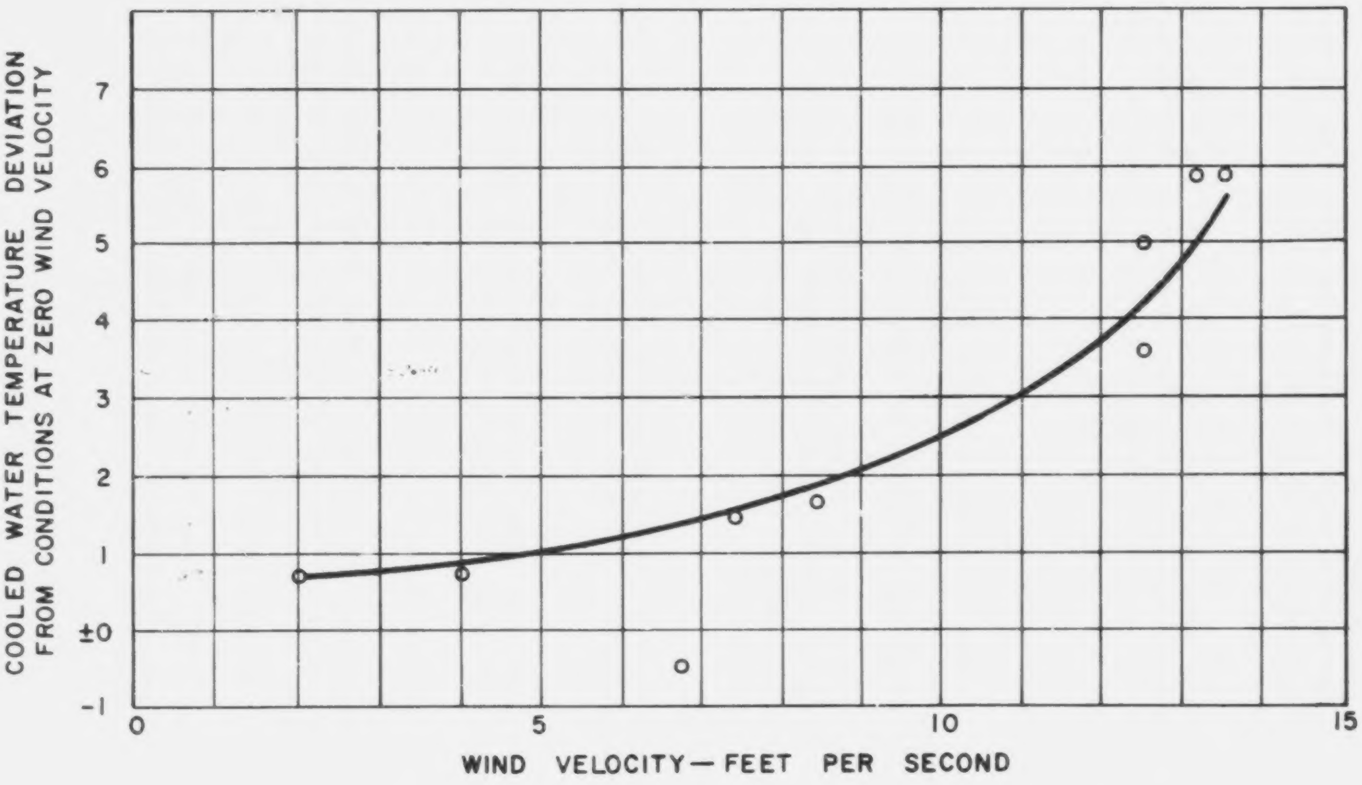


FIGURE A15—WIND EFFECT UPON NATURAL-DRAFT TOWER PERFORMANCE (4A)

zero wind velocity). Figure A16 shows test data taken at Ibbenbüren for 28 hours continuously. The bottom two curves show the marked effect of wind speed on the tower performance.

Since it rained occasionally during the testing periods, the observers were able to actually measure the effect of rain on the tower performance. Rain was found to worsen the cooling effect of the tower. This was attributed to the fact that rain cools the air inside the tower, reducing the thermal lift which results in a lower air flow across the coils. Table A-II shows operating results as logged by station operating personnel with five operating conditions selected at random.

Water-Side Chemistry

Because of lack of experience with aluminum in the condensate circuit and the effects of high purity condensate on the life of aluminum, Preussag conducted a series of laboratory tests before placing the dry-type cooling tower into operation. During the laboratory tests, with pH held from 7.0 to 9.0 it was observed that the aluminum content in the water became very high, ranging up to 4 mg per liter.

It was determined that the high solubility of the aluminum was a result of a brass pump in the test installation and the presence of copper ions caused the aluminum to dissolve. For that reason, the use of copper and copper alloy products was avoided in the thermal cycle and the cooling tower cycle of the 150-mw unit.

When the unit was first placed into service, the condensate pH was held between 8.5 and 8.7, but experience showed that when 8.5 pH was exceeded the solubility of aluminum became too high. Based upon that experience, condensate is controlled to a pH value of 7.8 to 8.0 by the addition of hydrazine. Aluminum content is held to 0.002 mg per liter with the pH at 7.8 to 8.0.

The expected life of the aluminum tubes is estimated to be from 20 to 30 years based upon the solubility now experienced.

In order to prevent aluminum in the condensate from reaching the boiler, the portion of the condensate which is returned to the thermal cycle is passed through special screening filters coated with asbestos, reducing aluminum content of the feedwater to 0.01 milligrams per liter. After the screen, the condensate to the boiler passes through a cation and anion demineralizer polisher where the aluminum content is further reduced to 0.002 milligrams per liter.

The oxygen content of the circulating water is from 0.1 to 0.3 milligrams per liter; the oxygen content in the thermal condensate is 0.01 to 0.02 milligrams per liter.

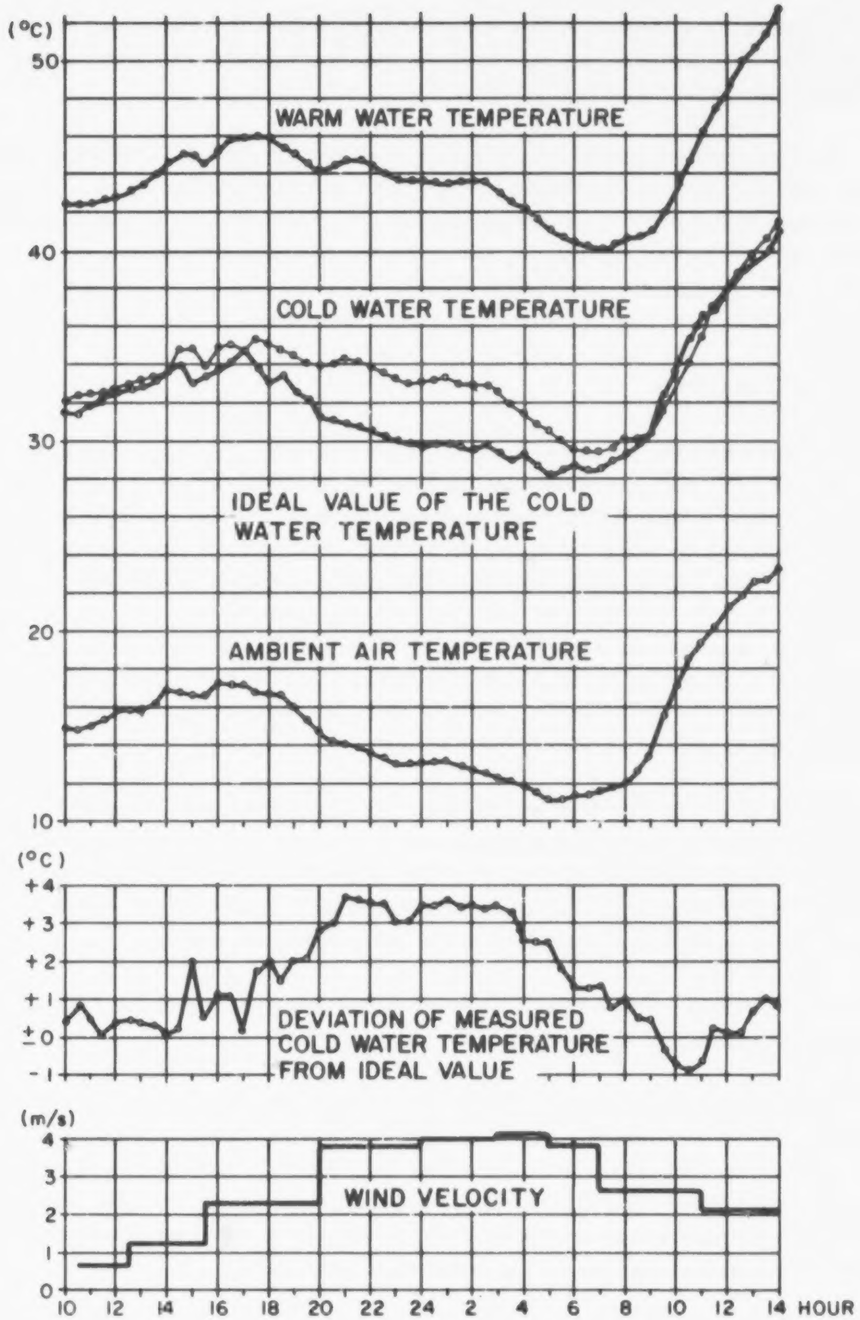


FIGURE A16— DRY-TYPE, NATURAL-DRAFT COOLING TOWER: IBBENBÜREN PLANT PERFORMANCE TEST RESULTS (4A)

TABLE A-II

Operating Data — Preussag-Kraftwerk
150-Mw Turbine-Generator — Ibbenbüren
with Natural-Draft Dry Cooling Tower

Ambient Air Temperature (°F)	Wind Velocity (mph)	Back Pressure (in. Hg)	Auxiliary Power for Pumps (mw)	Net Output (mw)
60.3	2.2	3.1	1.0	152.5
60.4	9.0	3.5	1.0	147.2
63.3	2.5	2.9	1.0	135.2
63.7	2.9	3.0	1.0	126.3
65.1	5.2	2.2	1.0	71.4

The above information supplied by the Stein Kohlenberg-Werke Ibbenbüren with back pressures calculated from the saturation water temperature.

After 1 year of operation, the unit was taken out of service and inspection of the circulating water system and the condenser did not reveal any pitting or corrosion.

Maintenance

After 2-1/2 years of operation, there is a slight coating of coal dust and soot on the external surfaces of the fins and tubes which is not considered to be of significance as far as adversely affecting performance. It is planned to remove the dirt coating with detergent and water on the next scheduled shutdown.

There have been no extraordinary maintenance problems associated with the dry tower.

Conclusion

Based upon the continuous operation of the 150-mw unit at Ibbenbüren, it can be concluded that the operation of the Heller-type tower at Ibbenbüren has been successful.

VOLKSWAGEN PLANT

Introduction

On December 4, 1969 John P. Rossie, accompanied by Mr. Hans-Bernd Gerz of GEA, visited the steam-electric generating plant of the Volkswagen manufacturing plant in Wolfsburg. During the visit, they interviewed Mr. F. Wehrberger, Plant Engineer for Utilities, and his assistant, Mr. Erich Kirchhübel.

At the time of the visit, all three 50-mw units equipped with dry towers were carrying rated load. The turbine back pressure was 1.2 inches Hg (0.04 atmospheres) with ambient air temperature approximately 36°F.

Description of Station

The power station Wolfsburg of the Volkswagenwerk AG plant in Wolfsburg, West Germany supplies all electrical power and steam for the automobile manufacturing processes and also to the Town of Wolfsburg, which is heated by the steam extracted from the plant turbines. Approximately 24,000 people are employed at the plant, and the Town has a population of approximately 85,000. Many of the plant workers live in neighboring towns.

The power plant is divided into two sections; the old section is equipped with evaporative-type cooling towers to cool condenser circulating water and the new section is equipped with direct condensing dry towers. The old section has five automatic-extraction type turbines, each of 8.5-mw capability for a total of 42.5 mw. The new section has three 48-mw units.

The reason for constructing the dry-type towers with the three 48-mw units rather than to continue the use of wet towers was the shortage of water at the plant. There was not enough water available for evaporative towers without bringing it in from a long distance at a high price.

Contrary to the dry tower installations at Rugeley and Ibbenbüren, the dry towers at the Volkswagen generating plant are the direct condensing type in which exhaust steam is conveyed from the turbines through large-diameter pipes and is condensed in the cooling coils. The V-W dry towers are of the mechanical-draft type, manufactured by GEA of Bochum, Germany.

Figure A17 shows the mounting of the direct, air-cooled condenser on the roof of the turbine house.

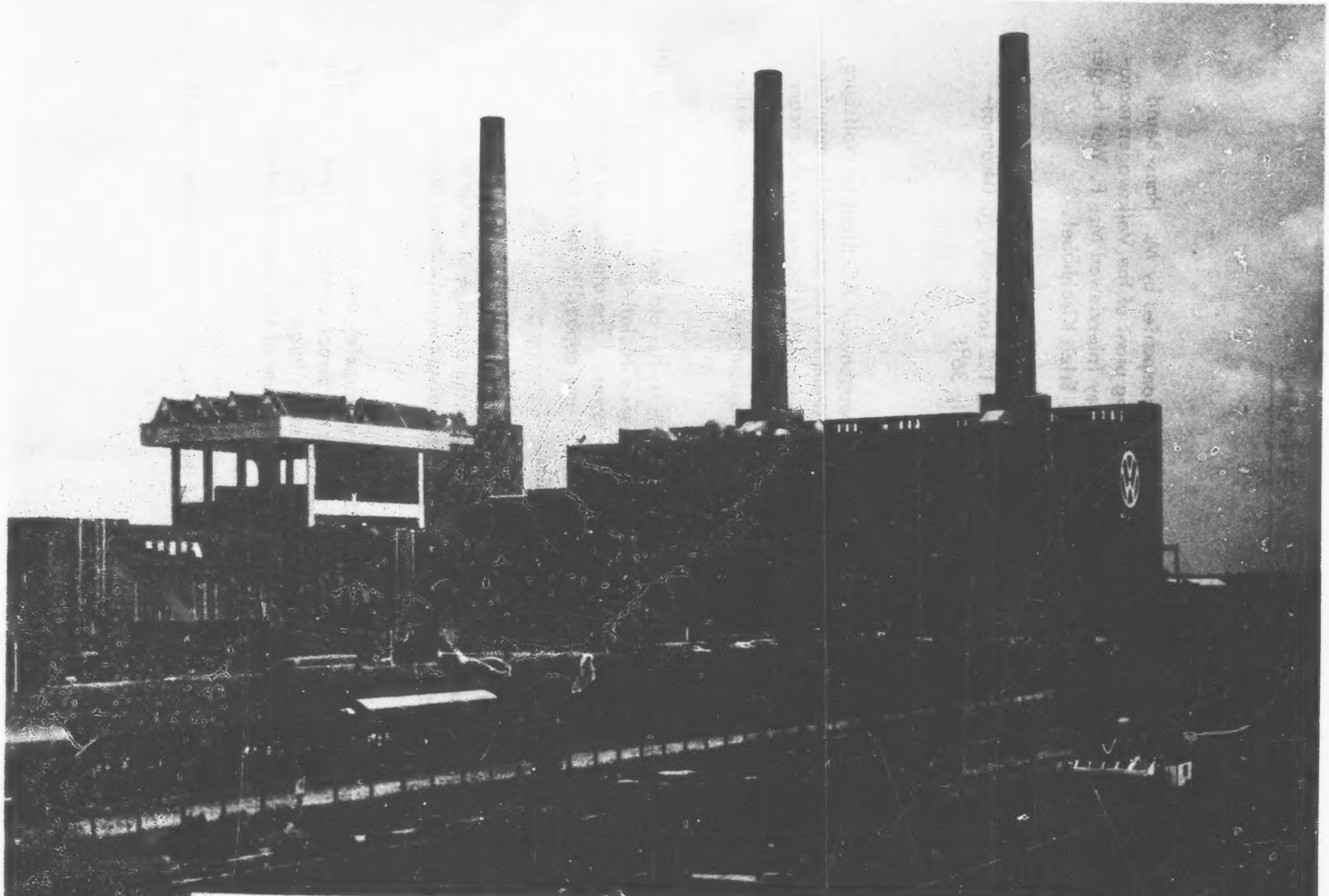


FIGURE A17 — VOLKSWAGEN PLANT WITH DIRECT-TYPE, AIR-COOLED CONDENSER UNITS ON PLANT ROOF (V-W PHOTO)

All the turbine-generators in the plant are of the automatic-extraction type which draw off steam from the turbines for processing and heating at a constant pressure over the varying turbine-load range. Since the demand for extracted steam is highest during the winter months because of the steam-heating load, the heat rejection duty of the cooling towers is lowest during the cold-weather months and highest during the warm months. The reason for this is that the amount of steam passing through the turbine to the condenser, for any given electrical load, varies with the amount of steam extracted for process. Consequently, the operating characteristics of the condensing system of an automatic-extraction type turbine-generator plant are quite different from the typical utility steam-electric generating plant where only enough steam for feedwater heating is extracted from the turbine, and the heat rejection load from the turbine exhaust steam is almost directly proportional to the electrical load on the unit. Other than the foregoing, the problems associated with the two types of plants are the same, and the experience of the V-W plant can be utilized by prospective purchasers of dry tower equipment.

Throttle steam conditions of the new section of the plant are 1,600 psi and 977°F. Three fuels—coal, natural gas, and residual oil—are burned in the plant.

The first 48-mw unit went into service in 1961, and the last unit in 1966.

Condensation Circuit

The exhaust steam from each of the 48-mw turbines is conveyed through pipes 10 feet in diameter to the air-cooled condenser units located above the turbine room. The air-cooling coils are arranged in the form of inverted V-shaped sections, similar to the deltas of the Heller system, except for the tube orientation and the position of the deltas or V-shaped tube bundles. Each of the three 50-mw generating units has its individual block of condensers, independent from the other two. Each 48-mw unit has 12 individual coils, and each coil is equipped with a 2-speed fan located beneath the coil, for a total of 12 fans per turbine unit.

Figure A18 shows a diagrammatic view of the piping from the turbine to the condenser. Note the motor-operated valves which can be operated remotely to take individual sections of cooling coils out of service in each block, while the remainder of the condensing coils remains in service. This feature is necessary for cold-weather operation, as described later.

Exhaust steam from the turbine condenses directly in the cooling coils and drains to a receiver where it is pumped back to the boiler. In contrast to the indirect system, no cooling water is used.

There are two different types of cooling coils in each block of condensers. One type is designated as the standard air-cooled condenser in which the steam

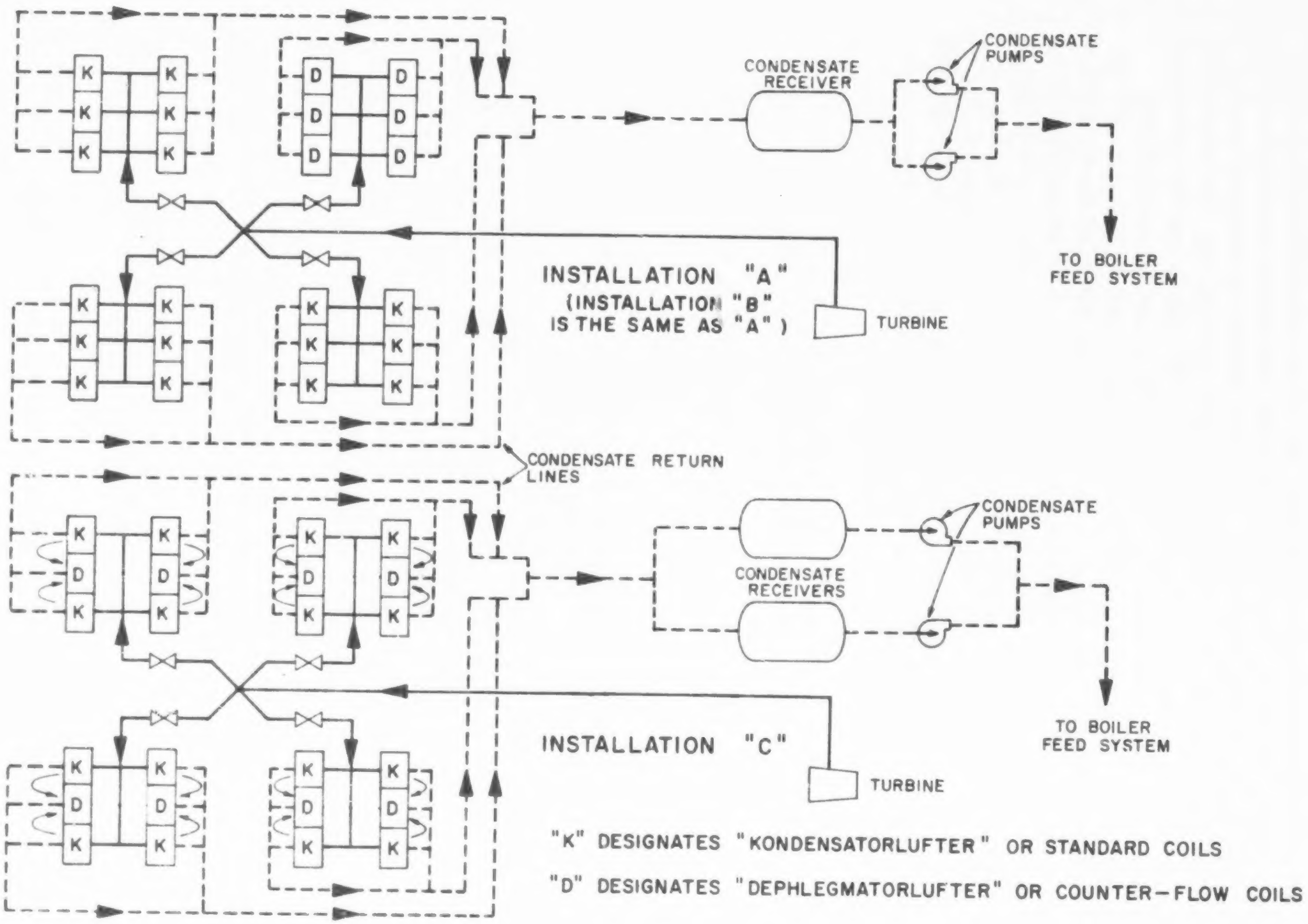


FIGURE A18—EXHAUST STEAM AND CONDENSATE PLANT OF AIR CONDENSING SYSTEM—VOLKSWAGEN PLANT

enters the coil from the top of the coil and is condensed as it travels downward along the sloping side of the V-section. In the standard condenser section, the flow of steam and of the water that forms during condensation is in the same direction; that is, from the top of the coil to the bottom. The other type of coil is designated as the counterflow condenser in which the steam distribution trunk is located along the bottom of the cooling-coil section so that the flow of steam into the coil is upward from the bottom to the top, while the flow of the condensate is downward.

The main purpose of the counterflow coil is to prevent the freezing of condensate during cold-weather operation, and also to prevent subcooling which results in a thermal loss to the turbine cycle. Although the counterflow coils have a lower heat transfer coefficient than the standard coil, it is necessary that a certain number of these be used during cold-weather operation.

In German, the standard coils are called "Kondensatorlufter" and the counterflow coils, "Dephlegmatorlufter". This designation is important in understanding the use of the operating diagram (Figure A19), which is explained on page 259.

The first two 48-mw units designated as "A" and "B", as can be seen from Figure A18, Groups A and B, each have three standard coil groups and one counterflow coil group, and each of the four groups can be shut off independently. Group C has a different arrangement in which there are four condenser groups, but each group consists of two standard coils and one counterflow coil.

The coils were designed and constructed by GEA with elliptical-shaped tubes and plate-type fins. The tubes and fins are made of steel, and protection against external corrosion and binding of the fin to the tube is accomplished by hot-dipped galvanizing.

Design Parameters

The turbine design back pressure at the V-W plant is 2.7 inches Hg at 59°F ambient temperature when condensing 242,000 pounds per hour of steam. This corresponds to an initial temperature difference (ITD) between saturated steam temperature in the condenser and the ambient air of 51°F, which is practically the same as the 50.5°F ITD at Ibbenbüren, and is representative of European design practice.

The average ambient temperature at Wolfsburg is 47°F; highest temperature, 91°F; and lowest temperature, −4°F.

The greatest condensing load is in the warm weather when there is less demand on the automatic-extraction type turbines. This trend can be seen from Table A-III, which was obtained from actual operating records of the V-W plant.

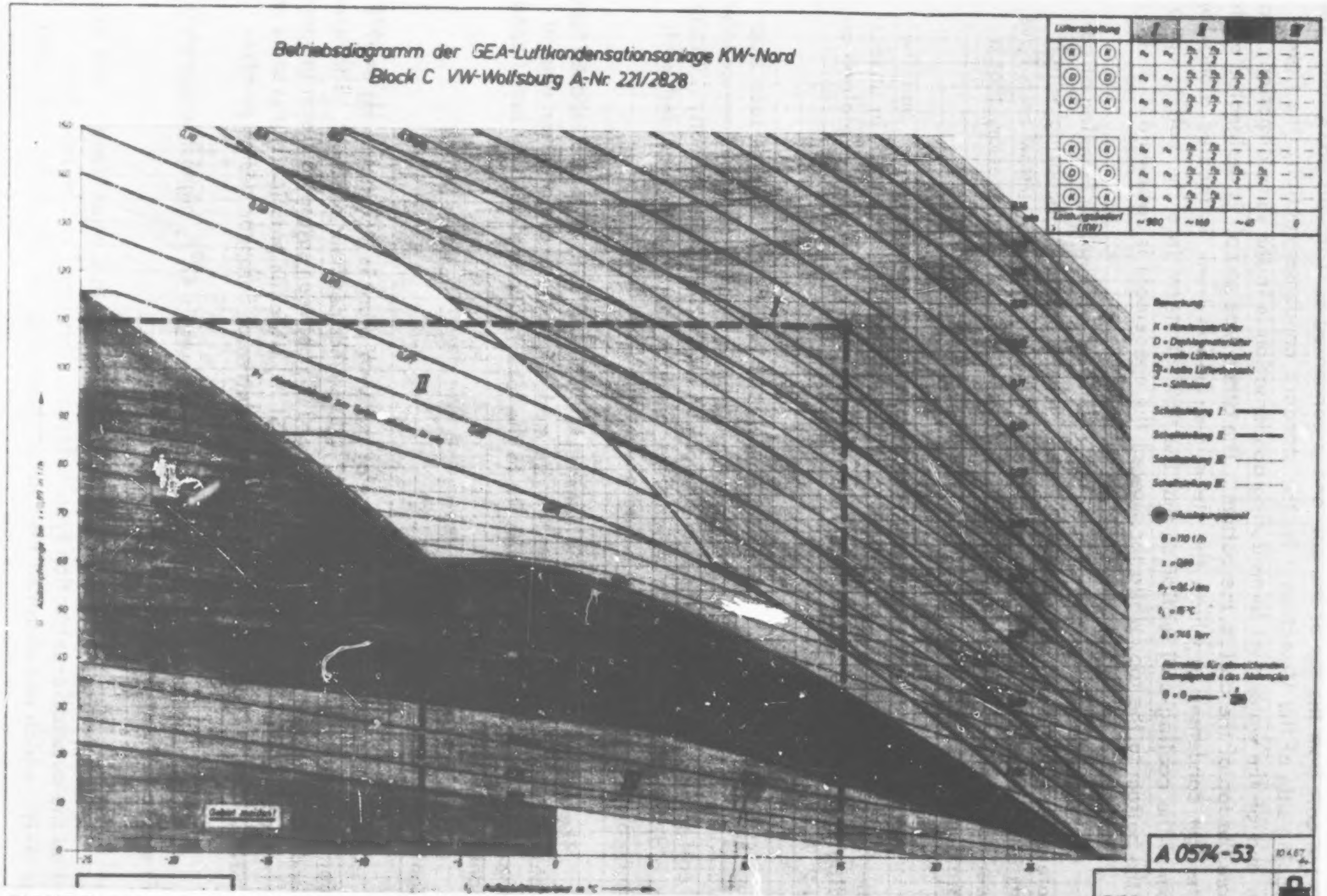


FIGURE A19—CALCULATED OPERATING CHARACTERISTICS (PREDICTED PERFORMANCE) FOR THE DIRECT AIR-COOLED CONDENSING SYSTEM—BLOCK 'C' OF VOLKSWAGEN PLANT (FROM GEA)

TABLE A-III

Operating Data — Power Station "Wolfsburg"
of the Volkswagenwerk AG.
49-Mw Automatic-Extraction Turbine-Generator
and Air-Cooled Condenser

Ambient Air Temp. ($^\circ F$)	Wind Vel. (mph)	Condenser Loading (lbs. of steam/hr.)	Back Pressure (in. Hg)	Auxiliary Power for Fans (kw)	Net Output (kw)	Number of Fans in Operation and Speed
82	7.82	198,414	4.20	850	30,000	12 (all fans) Full Speed
73	7.61	101,191	2.31	120	13,600	6 Half Speed
70	7.61	162,920	2.31	830	25,000	12 Full Speed
66	8.50	198,414	2.60	860	30,000	12 Full Speed
60	0.93	251,276	2.31	490	20,000	9 3 - Half Speed 6 - Full Speed
25	9.40	48,501	2.02	18	15,000	1 Half Speed
19	6.93	48,501	2.31	27	40,000	2 Half Speed
14	7.82	33,069	2.60	20	35,000	2 Half Speed
3	6.70	55,115	2.60	16	20,000	1 Half Speed

Table A-III also illustrates the great amount of operating flexibility available with the 12 fans and 2-speed motors.

An economic comparison of a wet tower and a dry-type cooling system was made before the final determination to construct the initial dry tower units (6A). Because of lack of space, a wet-type cooling tower would have had to be located approximately 3,000 feet from the power station in order to achieve a distance far enough from the coal storage pile to avoid problems of coal dust blowing into the cooling water.

The following assumptions were made in the analysis of the two towers:

	Seasons					
	Summer		Intermediate		Winter	
	Day	Night	Day	Night	Day	Night
Operating hours	1,785	735	1,428	588	1,071	441
Condensing load (metric tons/hr.)	110	73.3	70	46.7	20	20
Average air temperature, °C	15.8	12.2	7.1	4.9	1.2	0.1
Average relative humidity, %	77	84	82	87	85	87

Summer:	May to September
Intermediate Seasons:	March, April, October, November
Winter:	December to February
Capital Cost Basis:	12%
Cost of Power:	0.04 Deutschmarks per kwh (1.0¢ per kwh)
Cost of Water:	0.06 DM per ton (5.7¢ per 1,000 gallons)

A comparison of costs of two types of wet towers with direct, air-cooled condensing is as follows:

	Natural-Draft Wet Tower	Mechanical-Draft Wet Tower
Increased auxiliary power, DM/year	+61,600	+52,000
Cost of make-up water	-30,000	-24,000
Difference in maintenance costs	- 8,000	- 9,000
Difference in capital costs	-33,000	-57,000
Total cost savings, dry tower over wet tower, DM/year:	-10,000	-38,600

Note: + sign indicates penalty to dry tower;
- sign indicates credit to dry tower.

Based upon the analysis made by the Volkswagen AG engineers before the selection of the initial dry tower unit, an estimated annual saving of 10,000 DM ($2,500) favored the dry tower over a natural-draft wet tower, and 38,600 DM ($9,650) over a mechanical-draft wet tower.

Apparently, one of the significant economic factors in the selection of the direct condensing system at the Volkswagen plant was the great distance that a wet-type cooling tower would have had to be located from the plant because of space limitations.

Manpower Requirements of the Tower

There are no additional operators required to handle the air-cooled condensers. Instruments are installed in the central control room which enable the turbine control operator to oversee the tower operating conditions. All tower operating functions such as opening or closing valves to take cooling-coil sections out of service and fan speed changes are done from the control room. All operations are manually initiated from the control room and no automatic functions are provided for the air-cooled condenser.

Once each shift, an auxiliary operator checks the air-cooled condensers and the fan motors and gear box lubrication.

Freezing Problems

The method of preventing freezing of the coils during winter operation is to provide close control of the fan speed, number of fans and number of cooling-coil sections in operation for varying steam loads to the condenser, as hereinafter explained.

Freeze damage was experienced in A and B units after two winters of operation, when condensate pipes froze at the air aftercooler. This freezing was at probes which had been installed to determine air leaks. The probes were removed from A and B condensers and probes were not installed on the C condenser.

Condenser coils of installation A were frozen and damaged after three winters of operation at a time when air temperature was 10°F. Several tubes were frozen and four tubes were split, requiring replacement by welding in new sections. Freezing also occurred during the next winter in both A and B installations when the air temperature was 12°F. This time, a great many tubes were frozen, but only one tube was split, which required repair.

As a result of these freeze-ups, an extensive analysis of turbine and coil performance was made and certain conclusions were reached as to the reasons for the freezing. Improved operating methods were put into effect with the result that no further freezing damage was experienced.

The investigation disclosed that freezing of the cooling coils occurred under either of the following two conditions:

1. At a time when the extraction steam requirements were heavy and there was low steam flow to the condenser, a change in the electrical load or steam demand caused the turbine back pressure to rise because of greatly increased flow to the condensers.

   The rise in back pressure often was particularly sharp because of the fact that under the above conditions the counterflow section of either A or B installations was the only condensing surface in operation during light loads in cold weather. Although the counterflow sections have better characteristics with respect to prevention of freezing, the heat transfer is not as efficient as the standard coil sections because of the counterflow of steam and condensate.

   When the operators observed the rise in back pressure, they placed additional condensers into operation, with the effect that too much cooling was achieved and certain coil sections froze. The freezing problem was compounded by the fact that when freezing started, the turbine exhaust pressure increased because of the loss of condensing surface, and the operators often reacted by starting up additional fan capacity since the condensate temperature did not immediately indicate that freezing had occurred.

2. When minimum flow of cooling steam was flowing to the condenser, a condition common during cold weather and heavy extraction steam requirements, the exhaust steam was in a superheated state because it had been passed through the lower turbine blades for the purpose of cooling the blades and the expansion path of the steam, as traced on a Mollier Chart, was quite inefficient as compared to the condition where the exhaust steam has accomplished work in the turbine and had approximately 10 percent moisture when it reached the condenser. Since superheated steam has lower heat transfer characteristics than saturated steam, or steam with moisture content, any increase in the amount or temperature of cooling steam as a result of load changes, caused the turbine back pressure to rise and the operators to react by cutting in additional condensing surface or fan capacity, which often caused freezing.

To overcome the freezing problems, a number of air temperature probes were installed after the cooling coils and the operators were instructed to maintain 41°F when the air temperature reached 32°F. In order to accomplish this, it was necessary to keep the condenser loaded to approximately 90,000 to 100,000 pounds per hour of steam, and also to switch the fans on and off more frequently than had been done before. Subsequent tests have shown that the condenser steam loads can be reduced to one-half of the above figures without freezing.

Figure A19, which is the predicted performance of Block C (the latest installed 48-mw unit) was prepared by GEA; this figure shows the effect of air temperature, condenser steam load and fan operation on the turbine back pressure. The table in the upper right corner of Figure A19 shows recommended fan operation of the coil sections, which are shown in their arrangement by the designations:

K - Kondensatorlufter, standard coil, and

D - Dephlegmatorlufter, counterflow coil.

The Roman numerals I, II, III, and IV refer to the four zones shown on the curve in different shades of cross-hatching.

I - All 12 fans at full speed

II - All 12 fans at half speed

III - Four fans on parallel flow coils off;
2 fans on counterflow units at half speed

IV - All fans off

The guidelines illustrate two operating conditions:

1. Design point – 110,000 metric tons of steam per hour with 15°C ambient air and all fans in operation; resulting back pressure – 0.09 atmospheres (2.69 inches Hg).

2. 50,000 metric tons per hour steam to condenser, with –7°C ambient air and only the two counterflow fans in operation; resulting back pressure – 0.08 atmospheres (2.39 inches Hg).

The area in the lower left portion of the curve is to be avoided to prevent freezing.

The same performance curve applies for the condenser and turbine unit at part load with cooling sections out of service. With four of the sections, the steam loads as indicated would apply. With three sections in service and one out of service, the curve would apply when steam loads are 75 percent of those shown in the curve; that is, the 110 tons per hour would be equivalent to 82.5 tons per hour. With two sections in service, 50 percent of indicated steam load is used and with only one section in service, 25 percent of indicated steam load is used to read the curve. The foregoing explanation further illustrates the flexibility which the operators have in preventing freezing even during extremely cold weather and light condensing loads.

Start-up of condensing units during cold weather has not been a problem. The condenser is put into service with limited cooling coils operating and with fans off, and cooling coils and fans are brought into service as the condensing steam load is increased by following the operating performance curve.

The V-W plant has a peculiar winter operating problem in freeze prevention because the increase in extraction steam flow which occurs in the winter results in low turbine exhaust steam flow to the condenser.

Auxiliary Power Requirements

Each 48-mw turbine-generating unit has 12 two-speed motor-driven fans installed with the air-cooled condenser. The 12 fans for each 48-mw unit have a total full-speed power requirement of 860 to 890 kw. The half-speed fan requirements are 107 kw.

The fan power requirements are affected by the air temperature, since at colder temperatures the air density increases and more fan power is necessary.

At full speed with all fans in operation, the auxiliary power requirements are approximately 1.85 percent of output.

There are no circulating water pumping requirements associated with the direct condensing system.

Turbine Cycle Performance

Since the 48-mw turbines are automatic-extraction type, there is no generally accepted method of comparing the turbine cycle performance with a typical regenerative turbine cycle. However, the back pressure with 77°F air and full condensing load would be approximately 3.5 inches Hg, which is somewhat higher than the design of a typical wet tower installation. The higher back pressure results in a higher heat rate and it can reasonably be concluded that the turbine cycle heat rate for the V-W units equipped with air-cooled condensers is slightly higher than if they had been equipped with conventional wet towers.

One characteristic of the air-cooled condenser operation at the V-W plant which must be taken into account by the operators is the effect of the air in subcooling the condensate below the saturated temperature corresponding to the turbine back pressure. Because of the pressure drop in the exhaust steam trunk from the turbine to the air-cooled condenser, there is a steam-pressure drop which accounts for approximately 1 to 2°C (1.8 to 3.6°F) subcooling; this results in a slight thermal loss to the cycle. However, if close attention is not paid to operation of the fans when taking cooling-coil sections out of service with varying air temperatures and varying steam condenser loads, the subcooling effect can amount to from 10 to 12°C (18 to 22°F), which would have a marked effect on turbine efficiency.

Corrosion Problems

There have been no corrosion problems during the 8 years of operation of the air-cooled condensers, either on the exterior surfaces exposed to the atmosphere or to the internal surfaces which are in contact with steam and condensate.

Although the exhaust steam from the turbines contains 11 percent moisture, no erosion of tubes or piping has been reported.

Effect of Wind on Performance

Because the air-cooled condensers are equipped with mechanically driven fans, the effect of wind on the cooling tower performance is not a significant factor as it is in natural-draft towers. However, it is reported in (6A) that weather conditions have an influence on the condensing performance. Sunshine, cloudiness, rain and vapors from the wet cooling towers of the plant are reported to have a considerable influence on the cooling capacity of the direct condensers.

Water-Side Chemistry

The oxygen level in the condensate from the air-cooled condensers ranges from 0.005 to 0.007 mg per liter. The highest recorded oxygen has been 0.010 mg per liter.

During start-up, the oxygen content of the condensate is 0.06 to 0.067 mg per liter, but is quickly reduced by the air ejection equipment.

Hydrazine is used to control oxygen. Since the condenser tubes are steel, the V-W plant does not have any special problems in controlling a pH that is satisfactory for both aluminum and steel in contact with the condensate.

Although the entire cooling coil is under vacuum, no air leakage has been experienced, except for a leak during initial start-up, which was found weeks later to be in a condensate drain line under a pipe clamp and, consequently, very difficult to locate.

Maintenance

There have been no special maintenance problems associated with the air-cooled condensers. Normal maintenance is given to the fan gears and motors.

There have been no problems of dirt fouling the cooling-coil surfaces. The coils are cleaned once each year with water, under 200 psi pressure, and the time required to clean the condenser for one 48-mw unit is 3 to 4 hours. Before using high-pressure water for cleaning, a method of cleaning with compressed air was tried, but was abandoned for the water method. Since placing the condensing system into service, the lubrication system has been changed from the original design (6A). When first operated, summer- and winter-weight lubrications were used during the different seasons, but was changed to an all-weather weight when trouble in overloading the motor-driven lubricating oil pump was encountered during the intermediate seasons. The lubricating oil pumps are operated continuously since the idle fans rotate even when the motor drives are shut off, because of the wind effect through the coils, and thus require continuous lubrication. On the latest installation ("C") as a result of experience gained with A and B installations, the oil pump is driven directly by the fan gears so that lubrication is available whenever the fan blades are turning.

It was reported that maintenance requirements and expenses of the dry towers have been less than that for the wet towers installed in the old section of the power plant.

Conclusion

Although the extraction steam operation of the Volkswagen plant at Wolfsburg causes operational problems with respect to freezing during cold weather, a method of operation has been evolved which has resulted in successful winter operation.

Acceptance tests made indicated that the guaranteed vacuum was met with a 1°C margin of safety.

Keeping the coil surfaces free of dirt and preventing air leaks in the coils, which would impair operating efficiency, has not been a maintenance problem.

GYÖNGYÖS STATION

Introduction

On March 24, 1970 John P. Rossie, accompanied by Dr. L. Forgó of Hőterv and Mr. Istvan Lindner of Transelektro, visited the Gyöngyös steam-electric power station located at Gyöngyös, Hungary, approximately 60 miles east of Budapest. The Gyöngyös Station will have 600 mw of capacity served by dry towers when completed in 1972, and will be the largest station with dry towers to date. However, there is also under construction in Razdan, Soviet Armenia, a new station with three 200-mw units with dry-type towers, scheduled for completion in 1972.

At the time of the visit, one of the 100-mw units was in operation carrying approximately 40 mw (being limited because of ash conveyor operation) and the other 100-mw unit was out of service because of a feedwater heater leak, which was repaired quickly and the unit returned to service that day.

Description of Station

The initial units of the Gyöngyös Station went into operation in 1969, and additional generating units are currently under construction. Units 1 and 2, which were completed in 1969, are both 100 mw in capacity and are equipped with natural-draft, Heller-type dry towers. There are two 200-mw generating units under construction, one of which is equipped with a conventional wet tower utilizing mechanical draft and the other with a natural-draft Heller tower. A third 200-mw unit using a dry tower is planned to complete the station.

The Gyöngyös plant is a mine-mouth plant, located at the site of a newly opened strip-mining operation. Originally, all of the units, totalling 800-mw of generating capacity, were planned to be equipped with dry towers, but subsequent studies indicated that there was sufficient make-up water from the mining operation for a wet tower for one 200-mw unit. Plans were then changed and a conventional wet-type cooling tower and surface condensers were installed with one 200-mw unit.

There are approximately 150 million metric tons of lignite at the site, with heating value between 1,300 and 1,450 K calories per kg (2,340 to 2,620 Btu per pound); a moisture content of 33 to 34 percent; and an ash content of 22 to 30 percent.

Because of the extremely low quality of the coal, much difficulty has been encountered with the start-up of the plant, mainly in connection with the boilers and ash-handling systems. In view of the low-grade coal at the station, which is of much lower heating value than that which has been utilized in power generation heretofore anywhere in the world to our knowledge, such problems are to be expected.

At the time of the visit, many of the problems inherent to starting up a new plant had been solved and it appeared that the shakedown trouble would be overcome within a short time.

Figure A20 shows a photograph of the Gyöngyös Station with a view of the two 100-mw size reinforced-concrete towers. The station is of outdoor construction. Note that the concrete natural-draft towers serving the two 100-mw units are cylindrical in shape rather than hyperbolic.

Figure A21 shows the construction of the first 200-mw Heller tower, which was partially constructed at the time the photograph was taken. Note the slip-form construction of the concrete hyperbolic tower.

The two 100-mw steam turbines were manufactured in Hungary by Lang Engineering Works. The first 200-mw turbine will be a type L.M.2 manufactured in the USSR, and the second 200-mw turbine will be constructed in Hungary by Lang under a Brown Boveri Corporation license.

The generators are of Hungarian manufacture by the Ganz Electrical Works with water-cooled stator windings. The boilers are of outdoor design and were manufactured by the Hungarian Shipyards and Crane Factory.

The power plant is of modern design with centralized controls. Digital data logging is installed for continuous supervision of operation. The mining operations are quite extensive and lignite is delivered directly to the boiler bunkers by a conveyor system. The power output of the station is delivered to the Hungarian electrical grid over 120-kv and 220-kv transmission lines.

The turbines are covered by a thin-shelled reinforced-concrete building of half elliptical shape with telescoping sections which are equipped with wheels which run on separate parallel tracks so that the various sections of the turbine-generator can be uncovered for crane handling or dismantling.

Water Circuit

Figure A22 shows a diagrammatic arrangement of the circulating water circuit of the Heller system at the Gyöngyös Station. Approximately 42,000 gpm are circulated through the 100-mw unit towers and 93,000 gpm through the 200-mw unit towers.

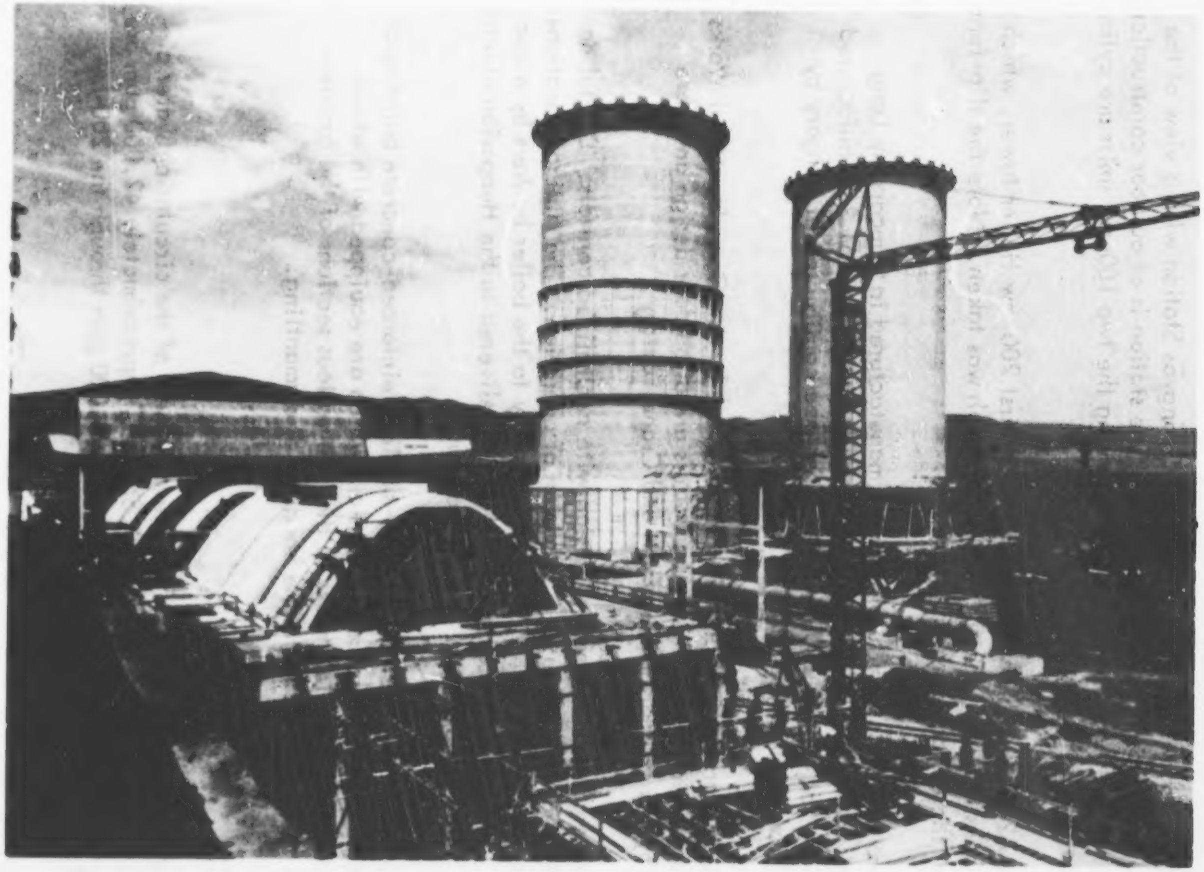


FIGURE A20— GYÖNGYÖS POWER STATION
TWO REINFORCED CONCRETE DRY-TYPE COOLING
TOWERS FOR 100 MW GENERATING UNITS (HÖTERV PHOTO)

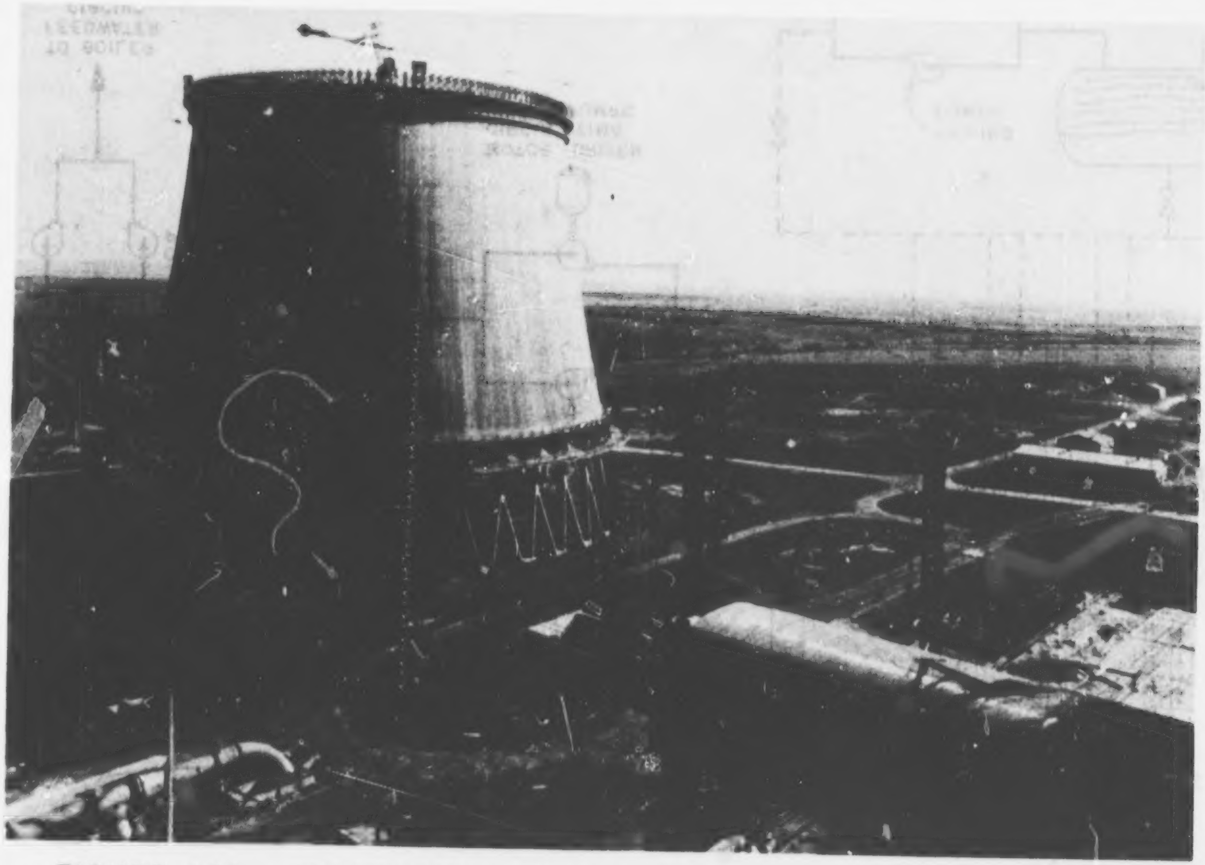


FIGURE A21 — REINFORCED CONCRETE TOWER FOR FIRST OF TWO 200 MW GENERATING UNITS IN THE GYÖNGYÖS POWER STATION(HÖTERV PHOTO)

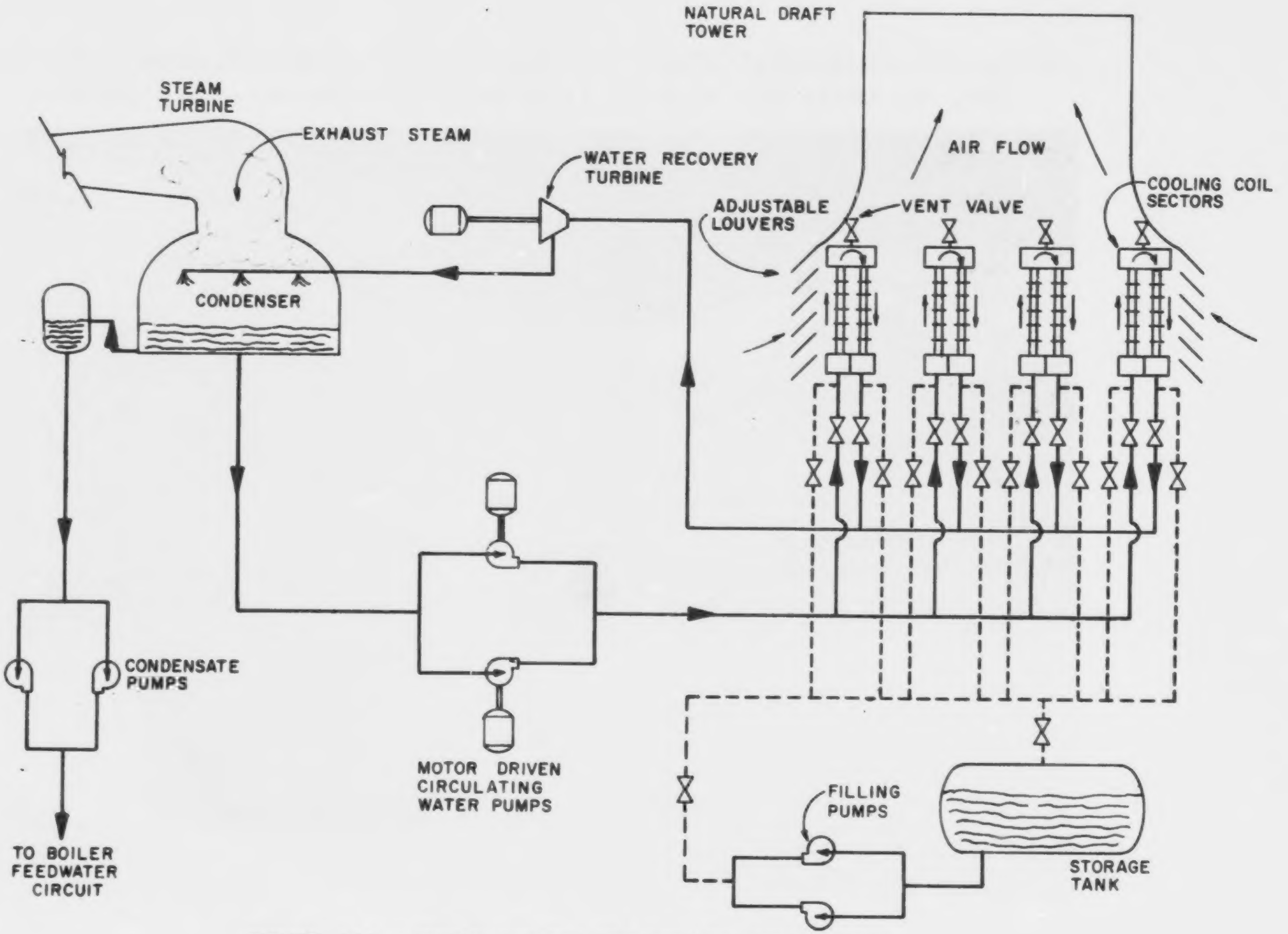


FIGURE A22 — WATER CIRCUIT FOR HELLER DRY TOWER GYÖNGYÖS STATION

Two half-capacity circulating water pumps are provided for circulating water to the tower and separate condensate pumps return the required amount of condensate to the boiler feedwater circuit.

The circulating water is carried to the dry towers through underground steel pipes and is directed into four cooling-coil sectors which can be individually drained or filled. The valves are of the butterfly type and are automatically controlled by the sequential tower control system.

An underwater circulating water storage tank and filling pumps are provided with each tower.

A head-recovery turbine with adjustable vanes is provided to convert the excess pressure head maintained in the cooling coils to electrical energy.

The auxiliary equipment of the units with dry towers is cooled by circulating water taken from the evaporative-type cooling tower serving the 200-mw unit.

Design Parameters

The design temperature difference between ambient air temperature and steam-condensing temperature of the dry-type cooling system at the Gyöngyös Station is 25.4°C, or 45.7°F for the two 100-mw units and 26°C, or 46.8°F for the 200-mw units. These design temperature differences compare with 35°F for the Rugeley tower and 50.5°F for the Ibbenburen tower. The air temperature range throughout the year is from approximately −10°F to 90°F.

The design heat rejection loads to the towers are 425 million and 900 million Btu per hour, respectively, for the 100-mw and 200-mw sizes.

The towers at Gyöngyös are all equipped with adjustable air louvers which are remotely controlled by the operators. The operating control mechanism of the louvers has a spring-loaded actuator between the driving motor and each individual louver so that the binding of one individual louver will not hinder the movement of the operating rod in its control of the remaining louvers served by that control motor. Dr. Heller advised that he considers louvers desirable with dry tower installations at any location where below-freezing temperatures are encountered.

The height of the 100-mw natural-draft towers is 367 feet with a base diameter of 176 feet. The height of the 200-mw natural-draft towers will be 380 feet with a base diameter of 357 feet.

The 100-mw dry towers each have 59 cooling deltas that are 15 meters in height, and the 200-mw towers have 119 deltas each. Since each delta consists of 6 individual Forgó coils, the 100-mw towers each have 354 cooling coils and the 200-mw towers have 714 coils.

The cooling coils are of aluminum and are of the same design and construction as the coils of the Ibbenbüren and Rugeley power stations.

Capital Costs of the Dry Tower

No figures are available as to the construction costs of the natural-draft, dry tower system at the Gyöngyös Station.

Manpower Requirements of the Tower

Since the tower filling and draining is completely automated, there are no additional manpower requirements for the dry-type cooling tower. Centralized control is provided with sufficient operating information available for the control operator to make all operating changes necessary to the tower from the control room.

Winter Operation

The start-up of the Gyöngyös Station afforded an excellent opportunity to observe the performance of a Heller-type tower under extremely adverse conditions caused by the starting and stopping of the generating unit frequently during freezing weather.

During the visit, it was reported that the two 100-mw units had been started up and taken out of service a total of approximately 80 times, and many of the starts and stops were during freezing weather.

The station director stated that none of the outages were caused by the dry towers and that no trouble was experienced in either draining or filling the towers during freezing weather. Automatic control provides rapid draining to the storage tank upon turbine shutdown in freezing weather and also provides automatic bypassing of the cooling sections during start-up in order to heat the entire charge of circulating water to a sufficiently high temperature to safely fill the coils.

Dr. Forgó explained the procedure for filling the cooling coils of the Heller system to insure that all air is removed from the empty coils during the filling process. Although each cooling-coil column is equipped with an automatic vent valve, trouble had been experienced in the past with air being trapped in certain coil sections and preventing water circulation, which lead to danger of freezing during cold weather.

In order to prevent freezing during the filling process, it is necessary that the coils be filled rapidly and that all air be vented from the coils. When the coils are filled from the inlet side of the tubes in the usual direction of water flow (upward in the inner three rows of tubes to the top water box where the flow direction is reversed downward through the outer three rows of tubes), the air is pushed out of the tubes ahead of the water into the top water box where most of the air is vented out through the automatic vent valve located on the top water box. However, some of the air may be carried over into the downflow tubes instead of being discharged through the vent, and may be trapped in the coil to prevent water circulating through individual tubes.

To prevent the air from being carried over into the downflow tubes, the following filling method has been developed and is used on the Gyöngyös towers:

> The delta sections are filled so that water enters the bottom of both the inlet and outlet headers; that is, water flows upward in the three tubes that normally carry water downward, and also upward in the three tubes which carry water upward during operation. During the filling process, the water in the three downflow tubes is maintained several inches higher than in the three upflow tubes so that the two flow sections of the coil are filled simultaneously with the downflow section leading in level. The water level difference is automatically controlled by the recovery turbine vanes. The purpose of this filling procedure is to insure that all air from the downflow side is either vented directly from the top water box, or into the few inches of air space ahead of the rising water in the upflow tubes. The rising water level in the upflow tubes pushes the air into the water box and out the vent. The air cannot re-enter the downflow tubes since they are filled with water and are sealed off to entry of air.

Dr. Forgó also explained that, although drainage of the coils must be accomplished rapidly during cold weather, the rate of drainage must be controlled, rather than to permit the 45-foot-high columns to drain as rapidly as free flow with the top vent and drain valve open would permit. It has been found by experience that too rapid drainage of the columns during freezing weather causes the water column in the tubes to break, slowing down the water flow and permitting the water in the coil to freeze into ice crystals inside the tube. This freezing, in itself, was not harmful to the tubes, and, since no damage was evident, the freezing was undetected until the tower was filled and placed into operation. The ice crystals from the draining operations caused restriction of water flow, and serious freeze-up occurred as soon as the tower was placed into operation.

Experiments were made to determine an acceptable draining velocity and the rate of drainage is now automatically controlled. Drainage of cooling coils at the Gyöngyös Station is accomplished in 1 to 1-1/2 minutes during cold weather.

Turbine Cycle Performance

Turbine cycle heat rate is reported to be 2,030 K calories per kwh for the 100-mw unit and 1,900 K calories per kwh for the 200-mw unit equipped with dry towers. This is equivalent to 8,050 Btu per kwh and 7,520 Btu per kwh, respectively. The turbine cycle heat rate of the 200-mw unit served by the evaporative tower and surface condenser is reported to be 1,960 K calories per kwh, or 7,750 Btu per kwh.

All turbine units are reheat type and will have six feedwater heaters in the cycle with final feedwater temperature of 446°F for the 100-mw units, 467°F for the 200-mw units served by dry towers, and 446°F for the 200-mw unit on the conventional tower.

It is noted that the throttle steam conditions of the 200-mw units with the dry towers are different than the steam conditions of the 200-mw units with conventional tower and surface condenser. Listed below are the turbine cycle conditions:

Gyöngyös Station Design Conditions

	Dry Tower Unit		Conventional Tower Unit
	100 mw	200 mw	200 mw
Number of units	2	2	1
Throttle steam pressure, psi	1,850	2,350	1,850
Steam temperature, °F	995/995	1004/1004	1058/1058
Final feedwater temperature, °F	446°	468°	446°

Since the plant has not been completed nor have the 100-mw units been placed into full operation, no comparison can be made as to operating results.

Corrosion Problems

The location of the Gyöngyös Station is on the Hungarian plains where the weather is generally dry. No corrosion problems are expected to be encountered in the operation. The initial units have been in operation less than 1 year, so no definite conclusion can be drawn; but, to date, the dirt and coal dust have not been a problem nor is there indication that they will be.

Conclusion

The use of the Heller-type dry tower with the Gyöngyös Station made it possible for the Hungarian Civil Construction Enterprise to make use of large lignite deposits for electrical power generation which otherwise could not have been used because of lack of cooling water for evaporative-type towers. The low calorific value of the lignite at Gyöngyös made it infeasible to transport the coal to a plant site where cooling tower make-up water was available. Thus, the coal resources would have remained unavailable without the use of the dry towers.

NEIL SIMPSON STATION

Introduction

On May 21, 1970 Edward A. Cecil and Clarence J. Steiert visited the Neil Simpson power plant of the Black Hills Power and Light Company at Wyodak, Wyoming, located about 6 miles east of Gillette, Wyoming, at an elevation of 4,600 feet above sea level. They were escorted on a tour of the plant by Mr. William Craig, Assistant Plant Superintendent. Because of the low fuel cost associated with the Company's coal mine, all four machines at this generating plant are operated at base load.

Description of Station

The plant is located at a coal mine owned by the Company and operated by a subsidiary, the Wyodak Resources Development Corporation. The plant consists of four small generating units. The first two units, rated 1,500 and 2,000 kw, respectively, are small conventional machines with standard condensers cooled by evaporative-type cooling towers. Unit No. 3 is an old 3,000-kw, 450-psig, 750°F condensing turbine-generator set which was moved from a retired plant to Wyodak for the purpose of experimenting with an air-cooled condensing system at the Wyodak coal mine site. Figure A23 shows the air-cooled condenser with louvers open. No additional water is available at this plant site, but there is an abundance of low-cost coal in a seam 70 to 90 feet thick with an overburden ranging generally in depth from 5 to 20 feet.

After a number of years of successful operation with air-cooled condensation by the experimental unit, the Company installed Unit No. 4, a 20-mw turbine-generator set, utilizing a direct, air-cooled steam condenser supplied by GEA of Germany. Figure A24 presents a view along the side of one of the A-frame condenser units. The turbine is rated at 20,180 kw nominal with 850 psi, 900°F steam. A 72-inch pipe conducts the exhaust steam from the turbine flange to two A-frame air-cooled condensing units mounted adjacent to the turbine room. The exhaust steam enters the condensing sections from the top and passes once through to the bottom of the heat exchanger where the condensate is collected in a header and flows by gravity to a collecting tank. The air passes over the finned tubes by cross-flow. The tubes, because of rugged design, are protected only by coarse hail screens. There are six fans, each of which is driven by two motors through a special gear reducer. The large motor is rated 150 horsepower constant speed for high-speed operation and the small motor is a two-winding unit requiring approximately 45 horsepower for half-speed fan operation (full-speed motor operation) and 10 horsepower for quarter-speed fan operation (half-speed motor operation). The fans are 20.8 feet in diameter and have six blades each. The fan speed is varied to provide the required turbine back pressure.

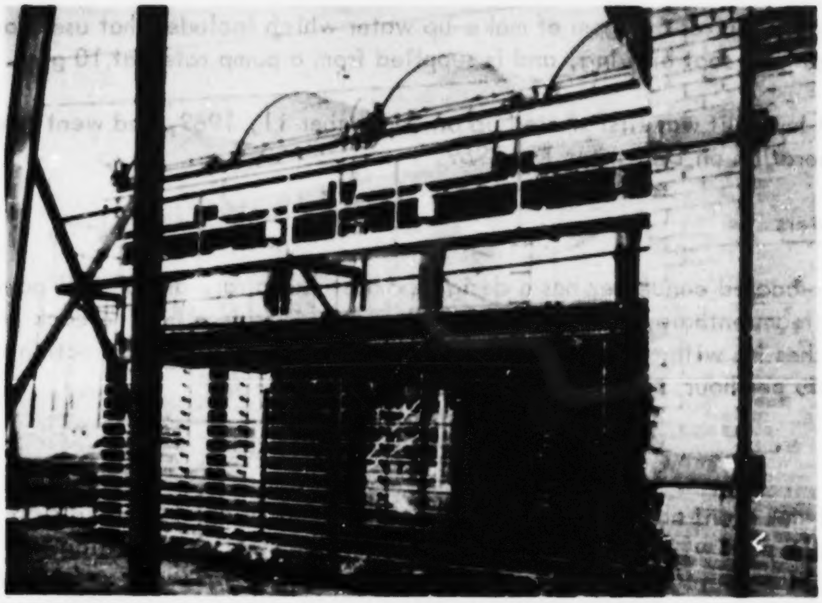


FIGURE A23—3,000 KW PILOT PLANT DIRECT-TYPE, AIR-COOLED CONDENSER INSTALLATION-NEIL SIMPSON PLANT, WYODAK, WYOMING

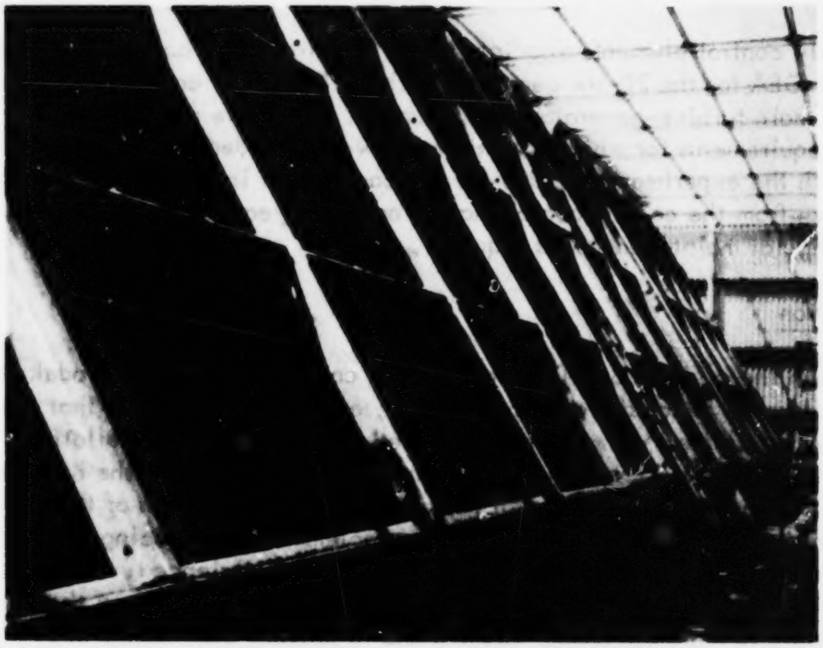


FIGURE A24—SIDE VIEW OF A-FRAME, DIRECT-TYPE AIR-COOLED CONDENSING UNIT—20 MW GENERATING UNIT, NEIL SIMPSON PLANT, WYODAK, WYOMING

The plant requires 5.7 gpm of make-up water which includes that used for boiler blowdown and soot blowing, and is supplied from a pump rated at 10 gpm.

The 20-mw unit was first started up on September 11, 1969, and went into commercial operation on December 15, 1969.

Design Parameters

The air-cooled condenser has a design exhaust steam rate of 167,301 pounds per hour with steam enthalpy of 1,027 Btu per pound to provide a turbine back pressure of 4.5 inches Hg with an ambient temperature of 75°F and a heat rejection rate of 155×10^6 Btu per hour. The design ITD is, therefore, 54.8°F.

Capital Cost

The 20-mw plant addition with an air-cooled condensing system costs approximately $315 per kw, with the air-cooled condensing system accounting for approximately 11 percent of the total. A standard cooling system with an evaporative-type tower would have cost approximately 3 percent less than the one that was used.

Manpower Requirements

The only control elements associated with the air-cooled condensing system as provided by GEA for the 20-mw generator unit are fan-motor control switches located at the main turbine-generator control board. There are no operating or maintenance requirements for which special manpower is needed. The Company's experience with the experimental 3,000-kw air-cooled unit indicates that dust should be blown from the coolers approximately once per year. Other than dust blowing, no regular maintenance was required.

Winter Operation

Although the original 3,000-kw air-cooled condensing unit at Wyodak was equipped with louvers for cold-weather operation, experience indicated that they were unnecessary. Louvers were therefore omitted from the 20-mw installation. Sidewalls shown in Figure A25 were provided around the periphery of the heat exchanger units to provide some measure of protection from the emission of the low stacks of other small units. To date, no serious problems due to freezing have occurred, although temperatures down to −33°F have been experienced. During severe cold-weather start-up, the operators alternately operate fans from the two condenser sections to maintain proper vacuum and to prevent coil freeze-up.

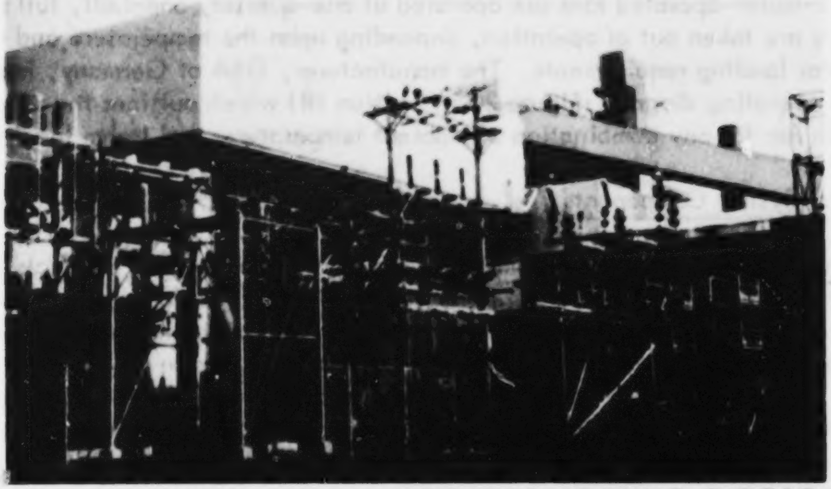


FIGURE A25 — SIDE WALLS ERECTED AROUND DIRECT-TYPE, AIR-COOLED CONDENSING UNIT-20 MW GENERATING UNIT, NEIL SIMPSON PLANT, WYODAK, WYOMING

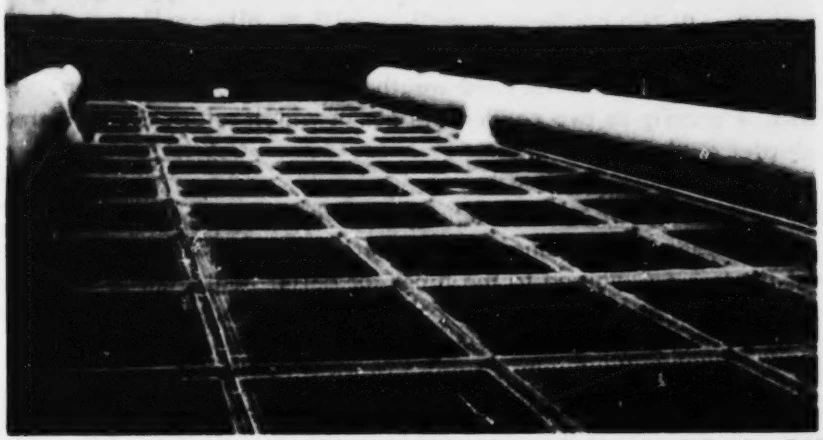


FIGURE A26 — STEAM HEADERS AND HAIL SCREENS-DIRECT-TYPE, AIR-COOLED CONDENSING UNIT-20 MW GENERATING UNIT, NEIL SIMPSON PLANT, WYODAK, WYOMING

The turbine, although chiefly a standard unit, is capable of quick start-up and quick shutdown, a characteristic which is important during severely cold weather.

The 2-motor-operated fans are operated at one-quarter, one-half, full speed and some fans are taken out of operation, depending upon the temperature and turbine-generator loading requirements. The manufacturer, GEA of Germany, has furnished an operating diagram (Figure 20 in Section III) which outlines the required speed of each fan for any combination of ambient temperatures and steam loading.

Description of System Components

Air-cooled condensation system. The GEA air-cooled system conducts the exhaust steam from the turbine through a 72-inch duct which branches under the air-cooled units to deliver steam to the top headers of the four condenser sections consisting of inclined heat exchanger finned-tube sections arranged in an inverted V-form similar in shape to the A-frame type of cottage construction. This arrangement is shown in Figure A26. Air is supplied to the center of this inverted V-arrangement by large fans, shown in Figure A27, similar to those used in conventional evaporative-type cooling towers. The steam enters the top headers of the cooling sections, passes down through the finned tubes and into the lower headers. From the lower headers, where the mixture now consists of steam and liquid condensate, the remaining steam and noncondensables pass upward through additional aftercooling sections which are provided with separate fans. The condensables are drawn off from the connecting header between the condensing and aftercooling sections. The noncondensables are drawn off from the top of the aftercooling sections.

Four condensing and two aftercooling sections, each with its own fan, are provided for the 20-mw unit. There are two rows of inverted V-type cooling sections, each consisting of two condensing sections with an aftercooling section in the middle. The fan speeds are controlled individually to minimize power consumption and prevent freezing problems during cold-weather or light-load operation. The condensate flows by gravity to two condensate pumps which return the condensate to the boiler feedwater cycle.

Cooling coils. The GEA heat exchanger sections are fabricated from elliptical-shaped carbon steel tubes arranged in staggered rows for best air flow characteristics. Steel fins are galvanized to the tubes for better heat transfer and corrosion protection. Spacers at the four corners of the fins assist the steel fin collars, which also act as spacers at the tube itself, to provide construction rigidity.

Auxiliary power requirements. The maximum fan power requirement of the air-cooled condensing system at Wyodak is 816 horsepower. This condition occurs at full-load operation with high ambient temperatures. The fan power requirements reduce in stages down to a minimum with reduced load and/or cold-weather opera-

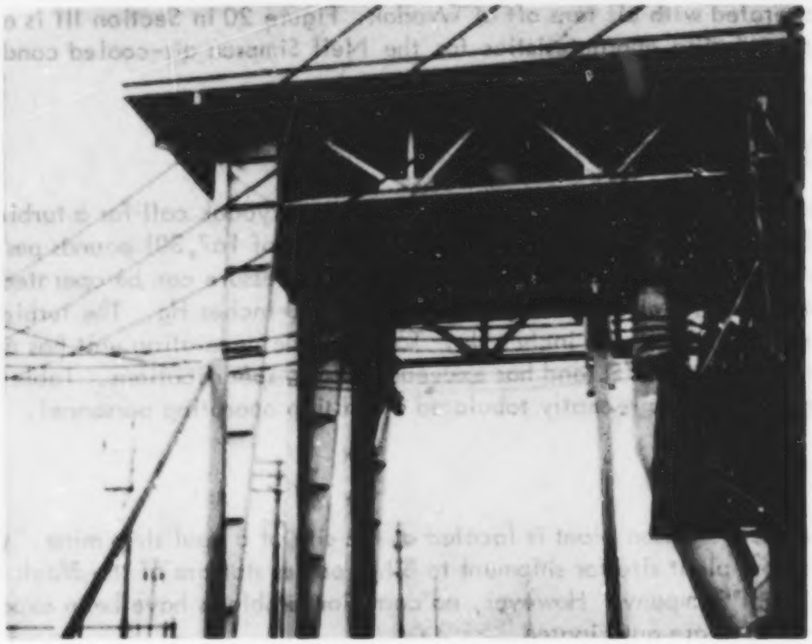


FIGURE A27— FAN ARRANGEMENT FOR DIRECT-TYPE, AIR-COOLED CONDENSING SYSTEM-20 MW GENERATING UNIT, NEIL SIMPSON PLANT, WYODAK, WYOMING

tion which could mean that all fans would be off. However, the operators as yet have not operated with all fans off at Wyodak. Figure 20 in Section III is a graph of calculated operating characteristics for the Neil Simpson air-cooled condensing plant.

Turbine Cycle Performance

The design conditions for the 20-mw unit at Wyodak call for a turbine back pressure of 4.5 inches Hg with an exhaust steam flow of 167,301 pounds per hour at an ambient temperature of 75°F. The turbine back pressure can be operated down to 2.0 inches Hg back pressure and up to as high as 7.0 inches Hg. The turbine trip is set at a back pressure of 7.5 inches Hg. To date, the generating unit has operated at temperatures up to 90°F, and has exceeded design specifications. Table A-IV provides operating data recently tabulated by station operating personnel.

Corrosion Problems

The Neil Simpson plant is located at the site of a coal strip mine. Coal is processed at the plant site for shipment to other power stations of the Black Hills Power and Light Company. However, no corrosion problems have been experienced to date, and none are anticipated.

Effect of Wind on Cooling Tower Performance

Wind has no apparent effect on the performance of this mechanical-draft, air-cooled condensing system except that due to low-stack emission from the smaller plant units when the wind is in a southwesterly direction. This effect is of a very minor nature and will not be present in any future plant construction.

Maintenance

Maintenance requirements for the air-cooled condensing system is practically nil and consists of normal fan-motor maintenance and air cleaning of the finned coils annually. No additional plant labor is required for this small maintenance. The air-cooled units are preferred by plant personnel over evaporative-type systems because of their small maintenance requirements. Plant water treatment costs are also very low for the air-cooled units.

Conclusion

The air-cooled condensing systems have been successfully utilized by the Black Hills Power and Light Company in Wyoming where a shortage of cooling water exists at the coal mine site. Both the original 3,000-kw experimental unit, which went into operation in the early sixties, and the 20-mw unit, started up in 1969,

TABLE A-IV

Operating Data — Neil Simpson Station
20-Mw Turbine-Generator with Mechanical-Draft,
Direct Air-Cooled Condensing System (6A)

Ambient Air Temperature (°F)	Gross Output (kw)	Steam Flow (lbs./hr.)	Condenser Loading (lbs./hr.)	Turbine Back Pressure (in. Hg)	Design Back Pressure (in. Hg)
16	20,687	203,760	149,920	4.33	6.2
30	20,479	204,000	149,400	4.38	3.7
44	19,604	190,000	144,000	4.22	4.8
52	20,400	200,000	147,000	4.26	4.5
4	20,600	206,000	152,000	4.71	5.2
43	17,500	172,000	130,000	4.14	4.1
- 8	19,667	186,200	146,160	5.56	5.3
-22	21,560	204,200	150,000	3.68	5.8
-18	21,267	203,333	150,667	4.76	6.4
46	20,600	205,000	145,000	2.70	3.6
26	22,600	211,000	156,000	3.41	3.5
74	22,000	205,000	153,000	3.99	3.99
80	21,600	203,000	153,000	4.68	4.45

have been operating at base load satisfactorily. The 3,000-kw unit enabled the operators to become acquainted with handling air-cooled systems and made the installation and start-up of the larger unit a routine matter. The Company has plans for a larger unit of approximately 150 mw when their system requirements or arrangements with neighboring systems will make a plant of this size feasible. The proposed larger station will be a few hundred yards from the present plant site so as to eliminate any adverse effects from the low-stack arrangements of the present plant site.

Appendix B

Engineering Weather Data

The air temperature data utilized in the economic optimization analyses were developed from information contained in the series of U. S. Weather Bureau publications entitled "Climatography of the United States No. 82, Decennial Census of United States Climate, Summary of Hourly Observations".

Economic optimization analyses were made for the 27 sites shown in Table A-V. The annual distribution of air temperatures for each of the 27 sites is summarized in Table A-VI.

TABLE A-V

Economic Optimization Analysis
Site Summary

Area	City	Site Number
Pacific:	Seattle, Washington	1
	San Francisco, California	2
	Los Angeles, California	3
Mountain:	Great Falls, Montana	4
	Boise, Idaho	5
	Casper, Wyoming	6
	Reno, Nevada	7
	Denver, Colorado	8
	Phoenix, Arizona	9
West North Central:	Bismarck, North Dakota	10
	Minneapolis, Minnesota	11
	Omaha, Nebraska	12
West South Central:	Little Rock, Arkansas	13
	Midland, Texas	14
	New Orleans, Louisiana	15
East North Central:	Green Bay, Wisconsin	16
	Grand Rapids, Michigan	17
	Detroit, Michigan	18
	Chicago, Illinois	19
East South Central:	Nashville, Tennessee	20
New England:	Burlington, Vermont	21
Mid-Atlantic:	Philadelphia, Pennsylvania	22
South-Atlantic:	Charleston, West Virginia	23
	Atlanta, Georgia	24
	Miami, Florida	25
Hawaii:	Honolulu, Hawaii	26
Alaska:	Anchorage, Alaska	27

TABLE A-VI

Annual Distribution of Air Temperatures

Site: No. 1, Seattle, Washington

Weather Station Location: Seattle-Tacoma Airport

Period: 1951-60

Air Temperature Range ($^{\circ}$F)	Percent of Time	Air Temperature Range ($^{\circ}$F)	Percent of Time
119/115	0	39/ 35	10.43
114/110	0	34/ 30	4.87
109/105	0	29/ 25	1.19
104/100	0	24/ 20	0.44
99/ 95	0.02	19/ 15	0.23
94/ 90	0.07	14/ 10	0.03
89/ 85	0.27	9/ 5	0
84/ 80	0.71	4/ 0	0
79/ 75	1.40	- 1/- 5	0
74/ 70	2.94	- 6/-10	0
69/ 65	5.11	-11/-15	0
64/ 60	8.56	-16/-20	0
59/ 55	14.51	-21/-25	0
54/ 50	16.68	-26/-30	0
49/ 45	16.48	-31/-35	0
44/ 40	16.06	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 2, San Francisco, California

Weather Station Location: International Airport

Period: 1951-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	1.13
114/110	0	34/ 30	0.11
109/105	0	29/ 25	0
104/100	0	24/ 20	0
99/ 95	0.01	19/ 15	0
94/ 90	0.06	14/ 10	0
89/ 85	0.17	9/ 5	0
84/ 80	0.46	4/ 0	0
79/ 75	1.13	- 1/- 5	0
74/ 70	3.25	- 6/-10	0
69/ 65	7.59	-11/-15	0
64/ 60	14.42	-16/-20	0
59/ 55	26.70	-21/-25	0
54/ 50	26.70	-26/-30	0
49/ 45	13.15	-31/-35	0
44/ 40	5.12	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 3, Los Angeles, California

Weather Station Location: International Airport

Period: 1951-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	0.11
114/110	0	34/ 30	0
109/105	0	29/ 25	0
104/100	0.01	24/ 20	0
99/ 95	0.05	19/ 15	0
94/ 90	0.08	14/ 10	0
89/ 85	0.32	9/ 5	0
84/ 80	1.34	4/ 0	0
79/ 75	4.33	- 1/- 5	0
74/ 70	10.05	- 6/-10	0
69/ 65	18.86	-11/-15	0
64/ 60	25.01	-16/-20	0
59/ 55	21.72	-21/-25	0
54/ 50	12.02	-26/-30	0
49/ 45	4.88	-31/-35	0
44/ 40	1.22	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 4, Great Falls, Montana

Weather Station Location: International Airport

Period: 1951-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	9.27
114/110	0	34/ 30	7.96
109/105	0	29/ 25	6.08
104/100	0.01	24/ 20	4.04
99/ 95	0.06	19/ 15	2.49
94/ 90	0.52	14/ 10	1.90
89/ 85	1.29	9/ 5	1.55
84/ 80	2.13	4/ 0	1.35
79/ 75	3.38	- 1/- 5	1.15
74/ 70	4.64	- 6/-10	0.78
69/ 65	5.93	-11/-15	0.58
64/ 60	7.25	-16/-20	0.49
59/ 55	8.60	-21/-25	0.17
54/ 50	9.37	-26/-30	0.05
49/ 45	9.47	-31/-35	0
44/ 40	9.49	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 5, Boise, Idaho

Weather Station Location: Boise Air Terminal

Period: 1951-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	10.01
114/110	0	34/ 30	9.46
109/105	0.01	29/ 25	5.95
104/100	0.09	24/ 20	3.50
99/ 95	0.50	19/ 15	1.69
94/ 90	1.55	14/ 10	0.61
89/ 85	2.62	9/ 5	0.30
84/ 80	3.60	4/ 0	0.16
79/ 75	4.28	- 1/- 5	0.07
74/ 70	5.61	- 6/-10	0.02
69/ 65	6.56	-11/-15	0
64/ 60	7.33	-16/-20	0
59/ 55	8.01	-21/-25	0
54/ 50	8.96	-26/-30	0
49/ 45	9.10	-31/-35	0
44/ 40	10.01	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 6, Casper, Wyoming

Weather Station Location: Casper Air Terminal

Period: 1956-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	9.48
114/110	0	34/ 30	9.19
109/105	0	29/ 25	7.79
104/100	0	24/ 20	5.64
99/ 95	0.04	19/ 15	3.69
94/ 90	0.75	14/ 10	2.28
89/ 85	2.29	9/ 5	1.32
84/ 80	3.23	4/ 0	0.83
79/ 75	3.96	- 1/- 5	0.51
74/ 70	4.82	- 6/-10	0.34
69/ 65	6.07	-11/-15	0.17
64/ 60	6.75	-16/-20	0.04
59/ 55	7.32	-21/-25	0.02
54/ 50	6.91	-26/-30	0
49/ 45	7.64	-31/-35	0
44/ 40	8.92	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 7, Reno, Nevada

Weather Station Location: Municipal Airport

Period: 1956-60

Air Temperature Range (^{o}F)	Percent of Time	Air Temperature Range (^{o}F)	Percent of Time
119/115	0	39/ 35	9.45
114/110	0	34/ 30	8.36
109/105	0	29/ 25	6.04
104/100	0.02	24/ 20	4.41
99/ 95	0.40	19/ 15	2.59
94/ 90	1.37	14/ 10	1.15
89/ 85	2.77	9/ 5	0.42
84/ 80	3.80	4/ 0	0.17
79/ 75	4.23	- 1/- 5	0.05
74/ 70	4.77	- 6/-10	0.01
69/ 65	5.44	-11/-15	0
64/ 60	6.52	-16/-20	0
59/ 55	7.87	-21/-25	0
54/ 50	9.64	-26/-30	0
49/ 45	10.37	-31/-35	0
44/ 40	10.15	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site:	No. 8, Denver, Colorado
Weather Station Location:	Stapleton Airfield
Period:	1951-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	8.18
114/110	0	34/ 30	8.22
109/105	0	29/ 25	6.31
104/100	0.01	24/ 20	4.09
99/ 95	0.11	19/ 15	2.46
94/ 90	1.18	14/ 10	1.36
89/ 85	2.69	9/ 5	0.89
84/ 80	3.79	4/ 0	0.41
79/ 75	4.98	- 1/- 5	0.25
74/ 70	6.26	- 6/-10	0.07
69/ 65	7.80	-11/-15	0.01
64/ 60	8.93	-16/-20	0
59/ 55	8.34	-21/-25	0.01
54/ 50	7.73	-26/-30	0
49/ 45	8.03	-31/-35	0
44/ 40	7.89	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 9, Phoenix, Arizona

Weather Station Location: Sky Harbor Airport

Period: 1951-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0.01	39/ 35	2.08
114/110	0.17	34/ 30	0.65
109/105	1.47	29/ 25	0.09
104/100	3.69	24/ 20	0
99/ 95	5.78	19/ 15	0
94/ 90	6.97	14/ 10	0
89/ 85	7.96	9/ 5	0
84/ 80	9.10	4/ 0	0
79/ 75	8.83	- 1/- 5	0
74/ 70	8.69	- 6/-10	0
69/ 65	8.85	-11/-15	0
64/ 60	8.75	-16/-20	0
59/ 55	8.77	-21/-25	0
54/ 50	7.52	-26/-30	0
49/ 45	6.16	-31/-35	0
44/ 40	4.46	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 10, Bismarck, North Dakota

Weather Station Location: Municipal Airport

Period: 1951-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	6.89
114/110	0	34/ 30	7.45
109/105	0.01	29/ 25	6.27
104/100	0.07	24/ 20	5.40
99/ 95	0.32	19/ 15	4.23
94/ 90	0.89	14/ 10	3.85
89/ 85	1.72	9/ 5	3.33
84/ 80	2.87	4/ 0	3.17
79/ 75	4.07	- 1/- 5	2.37
74/ 70	5.18	- 6/-10	1.49
69/ 65	6.46	-11/-15	0.88
64/ 60	7.00	-16/-20	0.54
59/ 55	6.91	-21/-25	0.23
54/ 50	6.42	-26/-30	0.10
49/ 45	5.93	-31/-35	0.03
44/ 40	5.91	-36/-40	0.01

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 11, Minneapolis, Minnesota

Weather Station Location: Minneapolis-St. Paul International Airport

Period: 1951-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	6.39
114/110	0	34/ 30	7.21
109/105	0	29/ 25	6.94
104/100	0	24/ 20	5.86
99/ 95	0.09	19/ 15	4.37
94/ 90	0.61	14/ 10	3.55
89/ 85	1.68	9/ 5	2.81
84/ 80	3.36	4/ 0	2.12
79/ 75	5.34	- 1/- 5	1.36
74/ 70	7.08	- 6/-10	0.71
69/ 65	7.87	-11/-15	0.35
64/ 60	7.93	-16/-20	0.11
59/ 55	6.86	-21/-25	0.05
54/ 50	6.13	-26/-30	0.02
49/ 45	5.50	-31/-35	0
44/ 40	5.70	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 12, Omaha, Nebraska

Weather Station Location: Eppley Airfield

Period: 1951-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	7.47
114/110	0	34/ 30	7.56
109/105	0.01	29/ 25	5.83
104/100	0.15	24/ 20	4.45
99/ 95	0.50	19/ 15	3.27
94/ 90	1.70	14/ 10	2.16
89/ 85	3.29	9/ 5	1.54
84/ 80	5.08	4/ 0	1.06
79/ 75	6.96	- 1/- 5	0.46
74/ 70	8.28	- 6/-10	0.17
69/ 65	8.22	-11/-15	0.03
64/ 60	6.91	-16/-20	0
59/ 55	6.37	-21/-25	0
54/ 50	6.15	-26/-30	0
49/ 45	6.19	-31/-35	0
44/ 40	6.19	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 13, Little Rock, Arkansas

Weather Station Location: Adams Field

Period: 1951-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	5.80
114/110	0	34/ 30	4.14
109/105	0.01	29/ 25	1.96
104/100	0.27	24/ 20	0.57
99/ 95	1.22	19/ 15	0.26
94/ 90	3.57	14/ 10	0.06
89/ 85	5.76	9/ 5	0.01
84/ 80	7.93	4/ 0	0
79/ 75	10.80	- 1/- 5	0
74/ 70	10.72	- 6/-10	0
69/ 65	9.16	-11/-15	0
64/ 60	8.30	-16/-20	0
59/ 55	7.66	-21/-25	0
54/ 50	7.27	-26/-30	0
49/ 45	7.63	-31/-35	0
44/ 40	6.90	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 14, Midland, Texas

Weather Station Location: Midland Air Terminal

Period: 1956-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	5.14
114/110	0	34/ 30	3.84
109/105	0.01	29/ 25	1.86
104/100	0.32	24/ 20	0.91
99/ 95	2.02	19/ 15	0.26
94/ 90	4.86	14/ 10	0.06
89/ 85	6.07	9/ 5	0.01
84/ 80	7.68	4/ 0	0
79/ 75	9.86	- 1/- 5	0
74/ 70	10.42	- 6/-10	0
69/ 65	9.04	-11/-15	0
64/ 60	8.21	-16/-20	0
59/ 55	7.73	-21/-25	0
54/ 50	7.20	-26/-30	0
49/ 45	7.63	-31/-35	0
44/ 40	6.87	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 15, New Orleans, Louisiana

Weather Station Location: Moisant International Airport

Period: 1951-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	1.46
114/110	0	34/ 30	0.54
109/105	0	29/ 25	0.10
104/100	0	24/ 20	0.02
99/ 95	0.14	19/ 15	0
94/ 90	2.61	14/ 10	0
89/ 85	7.07	9/ 5	0
84/ 80	11.17	4/ 0	0
79/ 75	19.06	- 1/- 5	0
74/ 70	13.56	- 6/-10	0
69/ 65	11.26	-11/-15	0
64/ 60	9.70	-16/-20	0
59/ 55	7.89	-21/-25	0
54/ 50	7.08	-26/-30	0
49/ 45	5.12	-31/-35	0
44/ 40	3.22	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site:	No. 16, Green Bay, Wisconsin
Weather Station Location:	Austin Straubel Airport
Period:	1956-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	7.40
114/110	0	34/ 30	9.35
109/105	0	29/ 25	7.85
104/100	0	24/ 20	5.87
99/ 95	0	19/ 15	4.26
94/ 90	0.10	14/ 10	3.66
89/ 85	0.75	9/ 5	2.64
84/ 80	2.01	4/ 0	1.83
79/ 75	3.78	- 1/- 5	1.08
74/ 70	5.40	- 6/-10	0.48
69/ 65	7.50	-11/-15	0.22
64/ 60	8.64	-16/-20	0.03
59/ 55	8.21	-21/-25	0
54/ 50	6.82	-26/-30	0
49/ 55	5.95	-31/-35	0
44/ 40	6.17	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 17, Grand Rapids, Michigan

Weather Station Location: Kent County Airport

Period: 1951-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	8.46
114/110	0	34/ 30	10.70
109/105	0	29/ 25	7.87
104/100	0	24/ 20	5.35
99/ 95	0.06	19/ 15	3.34
94/ 90	0.47	14/ 10	1.96
89/ 85	1.56	9/ 5	0.89
84/ 80	3.27	4/ 0	0.35
79/ 75	5.15	- 1/- 5	0.11
74/ 70	7.23	- 6/-10	0.01
69/ 65	8.42	-11/-15	0.01
64/ 60	8.12	-16/-20	0
59/ 55	7.37	-21/-25	0
54/ 50	6.52	-26/-30	0
49/ 45	6.45	-31/-35	0
44/ 40	6.33	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site:	No. 18, Detroit, Michigan
Weather Station Location:	City Airport
Period:	1951-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	9.22
114/110	0	34/ 30	10.08
109/105	0	29/ 25	7.05
104/100	0	24/ 20	4.30
99/ 95	0.01	19/ 15	2.83
94/ 90	0.54	14/ 10	1.49
89/ 85	1.69	9/ 5	0.70
84/ 80	3.58	4/ 0	0.19
79/ 75	5.88	- 1/- 5	0.05
74/ 70	8.22	- 6/-10	0.01
69/ 65	8.93	-11/-15	0
64/ 60	7.93	-16/-20	0
59/ 55	7.22	-21/-25	0
54/ 50	6.75	-26/-30	0
49/ 45	6.45	-31/-35	0
44/ 40	6.79	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 19, Chicago, Illinois

Weather Station Location: O' Hare International Airport

Period: 1956-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	8.41
114/110	0	34/ 30	9.80
109/105	0	29/ 25	7.29
104/100	0	24/ 20	4.25
99/ 95	0.02	19/ 15	2.53
94/ 90	0.57	14/ 10	1.96
89/ 85	1.85	9/ 5	1.30
84/ 80	3.79	4/ 0	0.91
79/ 75	5.61	- 1/- 5	0.52
74/ 70	8.28	- 6/-10	0.23
69/ 65	8.85	-11/-15	0.09
64/ 60	7.90	-16/-20	0
59/ 55	6.99	-21/-25	0
54/ 50	6.27	-26/-30	0
49/ 45	6.13	-31/-35	0
44/ 40	6.45	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 20, Nashville, Tennessee

Weather Station Location: Berry Field

Period: 1951-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	6.45
114/110	0	34/ 30	5.29
109/105	0.01	29/ 25	3.00
104/100	0.13	24/ 20	1.51
99/ 95	0.75	19/ 15	0.77
94/ 90	2.59	14/ 10	0.32
89/ 85	5.06	9/ 5	0.10
84/ 80	6.64	4/ 0	0.03
79/ 75	9.28	- 1/- 5	0.01
74/ 70	10.64	- 6/-10	0.01
69/ 65	9.55	-11/-15	0
64/ 60	8.41	-16/-20	0
59/ 55	7.97	-21/-25	0
54/ 50	7.26	-26/-30	0
49/ 45	7.06	-31/-35	0
44/ 40	7.16	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 21, Burlington, Vermont

Weather Station Location: Municipal Airport

Period: 1956-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	8.16
114/110	0	34/ 30	8.58
109/105	0	29/ 25	6.40
104/100	0	24/ 20	5.60
99/ 95	0.01	19/ 15	3.79
94/ 90	0.10	14/ 10	3.10
89/ 85	0.60	9/ 5	2.46
84/ 80	2.16	4/ 0	1.54
79/ 75	4.13	- 1/- 5	0.92
74/ 70	6.53	- 6/-10	0.44
69/ 65	7.64	-11/-15	0.19
64/ 60	8.02	-16/-20	0.06
59/ 55	7.91	-21/-25	0.02
54/ 50	7.47	-26/-30	0.01
49/ 45	6.88	-31/-35	0
44/ 40	7.28	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 22, Philadelphia, Pennsylvania

Weather Station Location: International Airport

Period: 1951-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	9.33
114/110	0	34/ 30	7.46
109/105	0	29/ 25	3.82
104/100	0.01	24/ 20	2.16
99/ 95	0.19	19/ 15	1.14
94/ 90	0.84	14/ 10	0.36
89/ 85	2.57	9/ 5	0.10
84/ 80	4.79	4/ 0	0
79/ 75	7.47	- 1/- 5	0
74/ 70	9.85	- 6/-10	0
69/ 65	9.22	-11/-15	0
64/ 60	8.38	-16/-20	0
59/ 55	8.10	-21/-25	0
54/ 50	7.56	-26/-30	0
49/ 45	8.00	-31/-35	0
44/ 40	8.65	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 23, Charleston, West Virginia

Weather Station Location: Kanawha Airport

Period: 1956-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	7.22
114/110	0	34/ 30	7.18
109/105	0	29/ 25	4.06
104/100	0	24/ 20	2.88
99/ 95	0.03	19/ 15	1.54
94/ 90	0.65	14/ 10	0.83
89/ 85	3.08	9/ 5	0.25
84/ 80	5.38	4/ 0	0.08
79/ 75	6.92	- 1/- 5	0.01
74/ 70	10.40	- 6/-10	0
69/ 65	10.82	-11/-15	0
64/ 60	8.75	-16/-20	0
59/ 55	7.85	-21/-25	0
54/ 50	7.54	-26/-30	0
49/ 45	7.60	-31/-35	0
44/ 40	6.93	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 24, Atlanta, Georgia

Weather Station Location: Municipal Airport

Period: 1951-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	5.34
114/110	0	34/ 30	3.09
109/105	0	29/ 25	1.28
104/100	0.02	24/ 20	0.50
99/ 95	0.33	19/ 15	0.22
94/ 90	2.02	14/ 10	0.09
89/ 85	4.57	9/ 5	0.02
84/ 80	7.13	4/ 0	0
79/ 75	10.07	- 1/- 5	0
74/ 70	13.52	- 6/-10	0
69/ 65	10.56	-11/-15	0
64/ 60	9.39	-16/-20	0
59/ 55	8.94	-21/-25	0
54/ 50	8.38	-26/-30	0
49/ 45	7.71	-31/-35	0
44/ 40	6.82	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 25, Miami, Florida

Weather Station Location: International Airport

Period: 1951-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	0.05
114/110	0	34/ 30	0
109/105	0	29/ 25	0
104/100	0	24/ 20	0
99/ 95	0.02	19/ 15	0
94/ 90	1.42	14/ 10	0
89/ 85	10.13	9/ 5	0
84/ 80	20.47	4/ 0	0
79/ 75	28.09	- 1/- 5	0
74/ 70	19.48	- 6/-10	0
69/ 65	9.24	-11/-15	0
64/ 60	5.15	-16/-20	0
59/ 55	3.16	-21/-25	0
54/ 50	1.68	-26/-30	0
49/ 45	0.81	-31/-35	0
44/ 40	0.30	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 26, Honolulu, Hawaii

Weather Station Location: International Airport

Period: 1951-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	0
114/110	0	34/ 30	0
109/105	0	29/ 25	0
104/100	0	24/ 20	0
99/ 95	0	19/ 15	0
94/ 90	0.01	14/ 10	0
89/ 85	1.34	9/ 5	0
84/ 80	17.70	4/ 0	0
79/ 75	39.44	- 1/- 5	0
74/ 70	31.81	- 6/-10	0
69/ 65	8.24	-11/-15	0
64/ 60	1.39	-16/-20	0
59/ 55	0.07	-21/-25	0
54/ 50	0	-26/-30	0
49/ 45	0	-31/-35	0
44/ 40	0	-36/-40	0

Total Percentage = 100.

TABLE A-VI (continued)

Site: No. 27, Anchorage, Alaska

Weather Station Location: International Airport

Period: 1956-60

Air Temperature Range (°F)	Percent of Time	Air Temperature Range (°F)	Percent of Time
119/115	0	39/ 35	8.13
114/110	0	34/ 30	9.48
109/105	0	29/ 25	9.06
104/100	0	24/ 20	7.55
99/ 95	0	19/ 15	6.39
94/ 90	0	14/ 10	4.28
89/ 85	0	9/ 5	3.25
84/ 80	0	4/ 0	2.41
79/ 75	0.09	- 1/- 5	1.42
74/ 70	0.57	- 6/-10	1.07
69/ 65	2.05	-11/-15	0.52
64/ 60	5.68	-16/-20	0.32
59/ 55	10.61	-21/-25	0.06
54/ 50	11.35	-26/-30	0
49/ 45	8.45	-31/-35	0
44/ 40	7.26	-36/-40	0

Total Percentage = 100.

Appendix C

General Specifications for Dry-Type Cooling System Applications

It is believed that dry-type cooling towers should be considered as part of the complete cooling and condensing system including pumps, fans (if mechanical draft), and condenser, in order that an economic evaluation may be made of the possible tower selection. Since the turbine-generator performance is most important in selecting the optimum dry-type tower for a particular plant, turbine-generator characteristics over the back pressure range expected for the various tower selections must also be considered.

The following factors should be a part of the economic analysis made to determine the size and type of the dry-type cooling systems.

1. Cooling tower capital cost versus ITD for the design heat rejection.

2. Fixed-charge rate. The components of fixed-charge rate are: interest or cost of money, depreciation, interim replacements, insurance and taxes.

3. Cost of fuel.

4. Operation and maintenance costs.

5. Differences in turbine-generator heat rates and capital costs for the various back pressure designs available.

6. Auxiliary power requirements of pumps and fans including head-recovery turbines.

7. Loss of turbine-generator capability during elevated ambient air temperatures.

8. Cost of replacing lost capability and energy. This cost would consider the capital investment, heat rate, fuel cost and operation and maintenance costs of the replacement capacity.

9. Air temperatures at the plant site.

10. Elevation of the site above sea level.

11. Time of system peak electrical load demand (winter or summer).

12. Annual operation pattern of the plant.

13. Comparison of mechanical-draft and natural-draft towers.

14. Range of cooling water temperatures.

15. Quantity of cooling air.

16. Method of evaluating ground-area requirements for towers.

After the heat load for the optimum ITD and the type of tower (mechanical-draft or natural-draft) have been determined, specifications should be drawn up to cover requirements. Included would be the following:

1. Wind loading and seismic design for the site.

2. Maximum noise level (for mechanical-draft towers).

3. Corrosion protection for cooling coils and fins, if required.

4. Material specifications for pumps, condenser, fan blades, and other components of the system.

5. Type of tower structure (concrete or structural steel), if natural draft.

6. Motor specifications (enclosures, voltages, type of insulation).

7. Means of modulation of air flow (louvers, fan speed changes, fan pitch variation or other available methods).

8. Extent of automation of operation and freeze protection desired.

9. Wind conditions at site (velocity and direction versus average hours per year).

10. Extent of shop assembly of components desired.

11. Capacity of cooling water storage and drain and refill provisions.

12. Hail protection required.

13. Means of removing air and noncondensable gases from system.

14. Allowable tower pumping head.

15. Provisions for handling coils and equipment at site during erection and maintenance periods.

16. Hydrostatic shop tests of cooling coils required.

17. Performance guarantees and tests to be made for acceptance of equipment after the plant is in operation.

Appendix D

Testing Upon Completion of Project

Although there is no accepted code as yet developed for testing the performance of a dry-type cooling tower installed with a steam-electric generating plant, the ASME Power Test Code for Atmospheric Water-Cooling Equipment, PTC-23-1958, will serve as a useful guide in establishing much of the required test procedure.

The main purpose of testing the dry tower installation would be to determine whether the guaranteed heat rejection can be achieved with the design ITD. In addition, information would be obtained during the test concerning the water-pressure drop through the cooling coils of the tower, the air-pressure drop across the cooling coils, the fan brake horsepower requirements, the effect of wind upon performance and the noise level of the fans.

It is important that performance tests be conducted during periods when the heat rejection load and the atmospheric conditions are stable. After reaching steady-state conditions, the duration of each test run should be at least 1 hour.

Since one of the most important aspects of the testing method is to obtain accurate data, a program to assure accuracy of measurements should be agreed upon and undertaken before starting the testing. Where the tests involve contractual obligations, the parties of the test should reach definite agreements covering the following:

1. Object of the tests.
2. The number of test runs.
3. Allowance for measurements and errors.
4. The method which will be used to operate the equipment.
5. The test apparatus to be used.

Provisions should be made to accurately measure the following:

1. The flow rate of circulating water.
2. The temperature of circulating water to and from the tower.

3. The ambient air temperature.

4. Power input to fans (for mechanical-draft towers).

The use of Pitot tubes for measurement of water flow is considered to be the most accurate measurement. Calibrated orifice plates or venturi tubes could also be used for water-flow measurement. The accuracy of temperature measurement should be within 1°F.

The following are suggested limitations for dry-type tower tests:

Flow rate of circulating water:	± 10 percent of design
Heat rejection:	± 20 percent of design
Cooling range of circulating water:	± 20 percent of design
Ambient air temperature:	± 10°F of design

The wind velocity during acceptance tests involving contractual obligations would generally be less than 10 mph.

During the test run, the variations from maximum to minimum would be held within the following limits of variation:

Circulating water flow:	5 percent
Heat rejection:	5 percent
Cooling range of circulating water:	5 percent

A time should be chosen for the test when the rate of change of the air temperature does not exceed 2°F per hour.

Readings should be taken at regular intervals not exceeding the following:

	No./hr.
Ambient air temperature:	6
Cold water temperature:	6
Warm water temperature:	6
Circulating water flow:	3
Wind direction and velocity:	6

Appendix E

Cooling System Cost Structure

Construction cost figures were developed for the various components of a dry-type cooling system from the turbine steam exhaust flanges through the cooling towers themselves. The requirements of various components were determined for initial temperature differences from 30°F to 80°F at 10° intervals. These values were then used in the computer program to determine construction costs at all intermediate points used in the analysis.

The steel towers used for natural-draft towers (Figure A28) were analyzed for 12 different sizes of towers. Cost estimates were prepared for cooling range from 0.4 to 0.6 of ITD which varied the assumed water flow rates and other parameters. Smooth curves were developed from the matrix of estimates representing optimum conditions so that the computer program could determine construction costs for all intermediate sizes.

An example of the cost structure used in our computer program for a dry-type, natural-draft cooling system for use with an 800-mw, fossil-fueled generating plant at sea-level elevation with a cooling system initial temperature difference of 60°F, a range of 30°F, a cooling water circulating requirement of 266,550 gpm, a tower height of 450 feet, a top diameter of 350 feet and a bottom diameter of 450 feet, based on 1970 price data, is as follows:

Cost Estimate of Natural-Draft Cooling Tower

A. Steel Tower with Aluminum Siding and Heat Exchangers:

1.	Central tower stack to include galvanized structural steel, aluminum siding, reinforced concrete footings, steel piles and excavation:	$ 1,633,000
2.	Bottom shed to accommodate heat exchangers and to include galvanized structural steel, aluminum roofing, reinforced concrete footings and excavation:	496,000
	Total Tower Structural Costs:	$ 2,129,000
3.	Heat exchangers:	$ 4,408,000
	Total Tower with Heat Exchangers:	$ 6,537,000

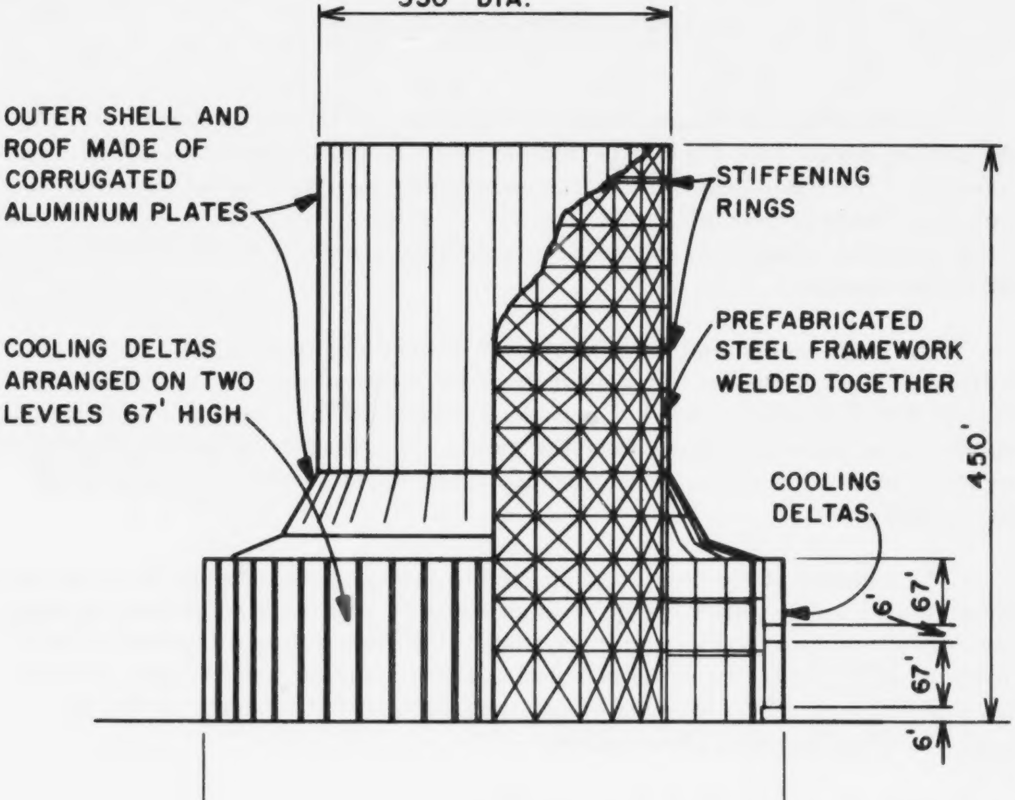

350' DIA.
OUTER SHELL AND ROOF MADE OF CORRUGATED ALUMINUM PLATES
STIFFENING RINGS
PREFABRICATED STEEL FRAMEWORK WELDED TOGETHER
COOLING DELTAS ARRANGED ON TWO LEVELS 67' HIGH
450'
COOLING DELTAS
6'
67'
67'
6'
450' DIA.

FIGURE A28 — OUTLINE OF NATURAL-DRAFT TOWER (FOR A 60° ITD DRY-TYPE COOLING SYSTEM FOR USE WITH AN 800 MW FOSSIL-FUELED GENERATING PLANT AT SEA LEVEL ELEVATION) USING STEEL AND ALUMINUM CONSTRUCTION

B.	Direct-Contact Condensers:	$ 832,000
C.	Piping, Valves, Pumps and Storage Tanks:	$ 3,381,000
D.	Controls and Automation applicable to Cooling System:	$ 500,000
	Complete Cooling System Construction Costs:	$11,250,000
	Plus 25% for Engineering, Contingencies, Interest and Taxes:	2,813,000
	Total Estimated Natural-Draft Cooling System Cost:	$14,063,000

Cost in $/kw of Plant Capacity: $17.60

Note: Above numbers rounded to nearest $1,000 and are based upon 1969-1970 cost levels.

For elevations other than sea level, cost curves were also developed for 3,000 feet and 6,000 feet, and the computer program was arranged to interpolate for elevations between these limits. For instance, a natural-draft cooling system for an elevation of 3,000 feet but otherwise the same as in the cost estimate above was priced at $14,705,000 (an approximate 4-1/2 percent increase over sea-level costs) and a system for an elevation of 6,000 feet was priced at $15,740,000 (an approximate 10 percent increase over sea-level costs).

Variations in the natural-draft tower size accounted for most of the cost difference at the various elevations used.

A comparison of tower dimensions for the same design conditions as for the detailed estimate shown above is shown below.

Dimensions of 800-Mw, Natural-Draft Tower for Fossil Fuel

Elevation Above MSL	Tower Height (ft.)	Top Diameter (ft.)	Bottom Diameter (ft.)
0	450	350	450
3,000	540	350	450
6,000	655	350	450

Costs for mechanical-draft towers were developed in a similar manner with the basic cooling unit consisting of heat exchangers, fans, gear boxes, motors, motor couplings, fan stacks, fan decks, supporting steel structure and concrete footings. Costs for this basic cooling unit were supplied by the Hudson Products Corporation. The number of cooling units were determined and all other accessories were added similar to that done for the natural-draft system. The cost figure used for a mechanical-draft, dry-type cooling system with a 60°F initial temperature difference and all other parameters the same as used for the sea-level unit for the natural-draft system is $13,281,000 and consists of the following components:

Cost Estimate of Mechanical-Draft Cooling Tower

Piping, Valves, Flanges and Tanks:	$ 1,427,000
Pumps and Recovery Turbines:	1,280,000
Controls and Automation applicable to Cooling System:	500,000
Direct-Contact Condenser:	832,000
Cooling Units:	6,586,000
Total of Above:	$10,625,000
Add 25% for Engineering, Contingencies, Interest and Taxes:	2,656,000
Total Estimated Mechanical-Draft Cooling System Cost:	$13,281,000

Cost in $/kw of Plant Capacity: $16.60

The cost of the mechanical-draft equipment is affected to a lesser degree by altitude than that for the natural-draft system. The total system cost for the mechanical-draft, dry-type cooling system with parameters as above at an elevation of 3,000 feet is approximately $13,580,000 (an increase of approximately 2-1/4 percent over sea-level costs) and at an elevation of 6,000 feet is approximately $13,945,000 (an increase of approximately 5 percent over sea-level costs).

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SECTION XV

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